OPTIMIZATION OF AN INTERNALLY FINNED ROTATING HEAT PIPE

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THESIS

OPTIMIZATION OF AN INTERNALLY
FINNED ROTATING HEAT PIPE

by

William A. Davis, Jr.

September 1980

Thesis Advisor: P. J. Marto

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With water as the working fluid, numerical results obtained for copper and stainless steel heat pipe condenser sections indicated

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Optimization of an Internally Finned Rotating Heat Pipe

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With water as the working fluid, numerical results obtained for copper and stainless steel heat pipe condenser sections indicated that for the maximum heat transfer rate, the designer should machine as many fins as the condenser material and the manufacturing process will allow. A saw tooth profile is preferable to spacing between fins.

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TABLE OF SYMBOLS

A	cross sectional area for flow in ft^2; finite element area
A_s	inside surface area of a smooth tube in ft^2
b (BVIN)	Height of the fin in ft; finite element factor
c	sonic speed in ft/sec; finite element factor
g	acceleration of gravity in ft/hr^2
h	convective heat transfer coefficient in Btu/hr-ft^2-°F
h_{fg} (HFG)	latent heat of vaporization in Btu/lbm
k_f (CF)	thermal conductivity of condensate film in Btu/hr-ft-°F
k_w (CW)	thermal conductivity of condenser wall in Btu/hr-ft-°F
L	finite element sides
M (AMTOT)	mass flow rate of condensate in lbm/hr
N	two-dimensional linear shape function
P	pressure of the vapor in lbf/ft^2
Q (QTOT)	heat transfer rate in Btu/hr
Q_s	heat transfer rate through a smooth tube in Btu/hr
R (RBASE)	internal radius of condenser in ft; thermal resistance in hr-°F/Btu
T (T)	temperature
U	velocity of liquid in ft/sec
X Y	axis of Cartesian system coordinate
x	coordinate measuring distance along the condenser length

y — coordinate measuring distance perpendicular to fin surface

z — coordinate measuring distance along fin surface

GREEK

α (FANGL) — fin half angle in degrees

δ^* (DEL) — condensate film thickness in ft

ε (EPS) — local trough width in ft

ϕ (PHI) — condenser cone half angle in degrees

ρ_e (RHOF) — density of the liquid in lbm/ft^3

ρ_v — density of the vapor in lbm/ft^3

σ — surface tension of the liquid in lbf/ft

μ (UF) — viscosity of the liquid in lbm/ft-sec

μ_v — viscosity of the vapor in lbm/ft-sec

ω (OMEGA) — angular velocity rad/sec

TABLE OF FORTRAN VARIABLE NAMES

ALFA(α)	FIN HALF ANGLE (RADIANS)
BFIN(b)	HEIGHT OF FIN (INCHES)
BVIN	HEIGHT OF FIN (FEET)
CALFA	COSINE OF ALFA
CANGL	CONE HALF ANGEL (DEGREES)
CBASE	INSIDE CIRCUMFERENCE OF CONDENSER (FEET)
CEXIT	INSIDE CIRCUMFERENCE AT CONDENSER EXIT (FEET)
CF (K_F)	THERMAL CONDUCTIVITY OF CONDENSATE FILM (BTU/HR·FT.°F)
CL	CONDENSER LENGTH (FEET)
CLI	CONDENSER LENGTH (INCHES)
CPHI	COSINE OF PHI
CRIT	CONVERGENCE CRITERION
DIV	FLOATING POINT VALUE OF NDIV
DMTOT	CONDENSATE MASS FLOW RATE
DOBF	NUMBER OF COLUMN WITHIN THE FIN
DOTH	NUMBER OF COLUMN WITHIN THE TROUGH
FANGL	FIN HALF ANGLE (DEGREES)
H	CONVECTIVE HEAT TRANSFER COEFFICIENT (BTU/HR·FT^2·°F)
HFG	LATENT HEAT OF VAPORIZATION (BTU/LBM)
IEL	THE ELEMENT NUMBER
JLC	NUMBER OF SYSTEM NODAL POINT LOCATED AT THE CENTER OF SYSTEM COORDINATE
JTC	NUMBER OF SYSTEM NODAL POINT LOCATED AT THE JUNCTION OF THE SYMMETRY BOUNDARY AND THE LINE OF INTERSECTION BETWEEN THE FIN AND THE CONDENSER WALL
KFF	NUMBER OF SYSTEM NODAL POINTS LOCATED ALONG THE FIN CONVECTIVE BOUNDARY
KFIN	NUMBER OF SYSTEM NODAL POINTS LOCATED ON THE SYMMETRIC BOUNDARY OF TRIANGULAR FIN SECTION NOT COUNTING POINTS AT BASE AND APEX
KT	NUMBER OF ROWS WITHIN THE WALL SECTION

NBAN	SYSTEM BAND WIDTH
NBOTF	LAST ELEMENT AT BOTTOM SIDE
NBOTI	FIRST ELEMENT AT BOTTOM SIDE
NDIV	NUMBER OF INCREMENT
NEFB	ELEMENT NUMBER AT BASE FIN
NEL	NUMBER OF ELEMENTS
NEST	ELEMENT NUMBER AT END OF TROUGH
NSNP	NUMBER OF SYSTEM NODAL POINTS
PHI(ϕ)	CONE HALF ANGLE (RADIANS)
PI(π)	PI
RBASE	INSIDE RADIUS OF CONDENSER BASE (FEET)
RBASEI(R)	INSIDE RADIUS OF CONDENSER BASE (INCHES)
REXIT	INSIDE RADIUS OF CONDENSER EXIT (FEET)
SALFA	SINE OF ALFA
SPHI	SINE OF PHI
THICK	CONDENSER WALL THICKNESS (FEET)
THICKI	CONDENSER WALL THICKNESS (INCHES)
TPHI	TANGENT OF PHI
ZFIN	NUMBER OF FINS
ZOA(c/a)	RATIO OF TROUGH WIDTH TO FIN BASE WIDTH

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I. INTRODUCTION

A. THE ROTATING HEAT PIPE

The rotating heat pipe is a closed container designed to transfer a large amount of heat in rotating machinery. Its three main component parts are: a cylindrical evaporator, a truncated cone condenser, and a working fluid as shown in Figure 1.

At rotation above the critical speed of a rotating heat pipe, the working fluid forms an annulus in the evaporator, and will be vaporized by heat addition to it. The vapor flows toward the condenser as a result of a pressure difference, transporting the latent heat of vaporization with it. External cooling of the condenser causes the vapor to condense on the inner wall and release its latent heat of evaporation. The centrifugal force due to the rotation has a component acting along the condenser wall that will act to drive the condensate back to the evaporator where the cycle is repeated.

In a conventional heat pipe, the force driving the condensate back to the evaporator is due to capillary action, which poses a limit to its operation. The rotating heat pipe is not limited by capillary action and, unlike the thermosyphon which depends on gravity to cause condensate return, can be used in any orientation [1].

B. OPERATING LIMITS OF A ROTATING HEAT PIPE

The first theoretical investigation of the rotating heat pipe conducted at the Naval Postgraduate School was performed by Ballback [2] in 1969. He studied the limitations of performance imposed on the rotating heat pipe due to various fluid dynamic mechanisms. Using existing theory and experimental correlations, he was able to estimate the sonic limit, boiling limit, entrainment limit, and the condensing limit of performance.

1. The Sonic Limit

When increasing the heat flux in a rotating heat pipe, it is possible to reach a limiting flow rate of the vapor brought on by a choked flow condition in the pipe. This condition imposes a limiting value on the amount of energy the vapor can transport, thus reducing the effectiveness of the heat pipe. The limiting heat transfer rate becomes

$$Q_t = \rho_v \, U_v \, A \, h_{fg} \tag{1}$$

and the vapor velocity is considered to be sonic,

$$U_v = c = \sqrt{g_0 kRT} \tag{2}$$

where

U_v = velocity of the vapor in ft/sec, and

A = cross sectional area for the vapor flow in ft^2

c = sonic velocity in ft/sec

g_0 = gravitational constant, 32.1739 ft-lbm/lbf-sec^2

k = ratio of specific heats

R = gas constant in ft-lbf/lbm °R, and

T = absolute temperature in °R.

2. Boiling Limit

Kutateladze [3] postulated that the transition from nucleate to film boiling is totally a hydrodynamic process. He determined a theoretical formula for predicting the burn-out flux

$$Q_t = K \sqrt{\rho_v}\, A_b\, h_{fg} \{\sigma\, g(\rho_f - \rho_v)\}^{1/4} \tag{3}$$

where

K = constant value

ρ_v = density of the vapor in lbm/ft^3

A_b = heat transfer area in the boiler in ft^2

h_{fg} = latent heat of vaporization in Btu/lbm

σ = surface tension in lbf/ft

g = acceleration of gravity in ft/hr^2

ρ_f = density of fluid in lbm/ft^3

ρ_v = density of vapor in lbm/ft^3

The experimental data obtained by Kutateladze suggested a value for K in the range of 0.13 to 0.19.

3. Entrainment Limit

The flooding constraint in a wickless heat pipe was examined by Sakhuja [4] who developed the correlation

$$Q_t = \frac{A_x\ C^2\ h_{fg}\sqrt{gD(\rho_f-\rho_v)\rho_v}}{\{1 + (\rho_v/\rho_f)^{1/4}\}^2} \tag{4}$$

where

Q_t = heat transfer rate in Btu/hr

A_x = flow area in ft^2

C = dimensionless constant, 0.725 for tube with sharp edged flange

h_{fg} = latent heat of vaporization in Btu/lbm

g = acceleration due to gravity in ft/hr^2

D = inside diameter of heat pipe in ft

ρ_f = density of the fluid in lbm/ft^3

ρ_v = density of the vapor in lbm/ft^3

4. Condensing Limit

Ballback [2] determined the condensation solution for a rotating heat pipe by modeling the condenser section of a rotating heat pipe as a rotating truncated cone. He developed the following expression for the condensation limit:

$$Q_t = \pi\left\{\frac{2}{3}\ \frac{k_f\ \rho_f\ \omega^2\ h_{fg}\{T_s-T_w\}^3}{\mu_f\ \sin^2\phi}\right\}^{\frac{1}{4}} \left\{[R_o + L\sin\phi]^{\frac{8}{3}}-R_o^{\frac{8}{3}}\right\}^{\frac{3}{4}} \tag{5}$$

where

Q_t = total heat transfer rate in Btu/hr

k_f = thermal conductivity of the condensate film in Btu/hr-ft-°F

ρ_f = density of fluid in lbm/ft^3

ω = angular velocity in 1/hr

h_{fg} = latent heat of vaporization in Btu/lbm

T_s = saturation temperature in °F

T_w = inside wall temperature in °F

μ_f = viscosity of fluid in lbm/ft-hr

ϕ = half cone angle in degrees

R_o = minimum wall radius in ft

L = length along the wall of the condenser in ft

μ_f = viscosity of the fluid in lbm/ft-sec

The condensing limit equation is a function of the geometry and speed of the rotating heat pipe, and the physical properties of the working fluid.

Tantrakul [5] calculated these limitations for a heat pipe with specific physical characteristics as shown in Table I, with the results shown in Figure 2.

TABLE I

Specification of a Typical Rotating Heat Pipe

Length	14.000	inches
Minimum diameter	2.000	inches
Wall thickness	0.125	inches
Internal half angle	1.000	degree
Rotating speed	2700	RPM

Obviously from the results in Figure 2, the condensing limit is the predominant limitation for the amount of heat that can be transferred from the heat pipe. However, the other limitations may become important as the heat pipe geometry and operating conditions are varied.

In order to augment the heat transfer capacity of the heat pipe, recent efforts have been aimed at raising the condensing limit line which may be accomplished by:

a. a high value of cone angle, to increase the centrifugal driving force,

b. some type of promoter of dropwise condensation to increase the value of the inside heat transfer coefficient, h, or

c. use of an internally finned condenser to increase the inner wall surface area and the value of h, since the presence of a fin will decrease the effective condensate film thickness.

A high value of cone angle means a departure from the cylinder or shaft shape. Since the principal known application for the rotating heat pipe is in the cooling of rotating machinery this approach was not pursued. Although effective promoters of dropwise condensation exist, none as yet can be considered to be permanent, and this approach was likewise ruled out. The remaining alternative, using internal fins in the condenser section to raise the condensing limit, was seen as the best choice.

C. ANALYSIS OF THE INTERNALLY FINNED ROTATING HEAT PIPE

Pursuing the addition of internal fins as a way to raise the condensing limit, Schafer [6] developed an analytical model for a heat pipe with a triangular fin profile as shown in Figure 3. He assumed one-dimensional heat conduction through the wall and fin. Corley [7] for this same case developed a two-dimensional heat conduction model using a Finite Element Method, and also assumed a parabolic temperature distribution along the fin surface. His results indicated a significant improvement in heat transfer performance of about 75% above that predicted by the one-dimensional model of Schafer [6]. However, Corley [7] cautiously noted a probable error of 50% existed at the fin apex, and consequently mentioned that there may be a total heat transfer error of as high as 15%. Tantrakul [5] modified Corley's computer program by increasing the number of finite elements in order to minimize the heat transfer error at the apex of the fin. His results with this modification converged with the results of Corley. Purnomo [1] developed a two-dimensional Finite Element Method solution using a linear triangular finite element model as shown in Figure 4. Purnomo's [1] Finite Element Method program also worked and converged. Purnomo's [1] code, when made to approach the geometry of a smooth tube, did not agree with the analytical and experimental data obtained by Schafer [6] for a smooth tube. This cast doubts about the validity of Purnomo's code.

The parametric studies conducted using Purnomo's code gave no clear indication of the best condenser geometry to maximize heat transfer. Also, his code was tedious to use and required numerous runs to obtain data since it was written to perform only one analysis at a time.

Clearly a computer program that could make numerous runs with minimal data input and could also automatically find improved designs would be valuable.

D. THESIS OBJECTIVES

The objectives of this thesis were therefore:

1. To modify Purnomo's [1] computer program so that it is compatible with the COPES/COMNIN program [8] and can be used for analysis and automated design of rotating heat pipes (internally finned or smooth).

2. To compare results using Purnomo's code with analytical results for a smooth tube obtained by Schafer [6] to determine if and where an error exists.

3. To use the sensitivity analysis capability of COPES/CONMIN in conjunction with the modified program to study heat transfer in an internally finned rotating heat pipe.

4. To use the resulting program to obtain an optimum design for an internally finned rotating heat pipe to obtain experimental data to compare with the analytical results.

5. To use the resulting program to obtain numerical results in place of data obtained from expensive experimental operations.

II. NUMERICAL OPTIMIZATION

A. BACKGROUND

Most design processes require the minimization or maximization of some parameter which may be called the design objective. For the design to be acceptable, it must satisfy a variety of physical, aesthetic, economic and, on occasion, political limitations which are referred to here as design constraints. While part of the design problem may not be easily quantified, most of the design criteria can be described in numerical terms.

To the extent that the problem can be stated in numerical terms, a computer program can be written to perform the necessary calculations. For this reason, computer analysis is commonplace in most engineering organizations. For example, in structural design the configuration, materials, and loads may be defined and a finite element analysis computer code is used to calculate stresses, deflections, and other response quantities of interest. If any of these parameters are not within prescribed bounds, the engineer may change the structural member sizes and rerun the program. The computer code therefore provides only the analysis of a proposed design, with the engineer making the actual design decisions. This approach to design, which may be called computer-aided design, is commonly used today.

Another common use of analysis codes is in tradeoff studies. For example, an aircraft trajectory analysis code may be run repetitively for several payloads, calculating the aircraft range, to determine the range-payload sensitivity.

A logical extension to computer-aided design is fully automated design, where the computer makes the actual design decisions, or performs trade-off studies with a minimum of man-machine interaction [9].

B. CONSTRAINED FUNCTION MINIMIZATION (CONMIN)

Vanderplaats [10] developed an optimization program CONMIN capable of optimizing a very wide class of engineering problems. CONMIN is a FORTRAN program, in subroutine form, that optimizes a multi-variable function subject to a set of inequality constraints based on Zoutendijk's [11] method of feasible directions [12].

Three basic definitions are required to discuss the use of CONMIN:

Design Variables - Those parameters which the optimization program is permitted to change in order to improve the design. Design variables appear only on the right hand side on an equation and are continuous.

Design Constraints - Any parameter which must not exceed specified bounds for the design to be acceptable. Design constraints may be linear or nonlinear, implicit or explicit, but they must be continuous functions of

the design variables. Design constraints appear only on the left side of the equations.

Objective Function - The parameter which is going to be minimized or maximized during the optimization process. The objective function may be linear or nonlinear, implicit or explicit, and must be a continuous function of the design variables. The objective function usually appears on the left side of an equation, but it may appear on the right side if it is also a design variable.

Design constraints and objective functions are usually interchangeable.

C. CONTROL PROGRAM FOR ENGINEERING SYNTHESIS (COPES)

Recall that the optimization program, CONMIN, was written in subroutine form. Vanderplaats [8] has developed a main program to simplify the use of CONMIN and to further aid in the optimization process. The user must supply an analysis subroutine named ANALIZ. What follows are programming guidelines to ensure compatability with COPES.

D. PROGRAMMING GUIDELINES

In developing any computer code for engineering analysis, it is prudent to write the code in such a way that it is easily coupled to a general synthesis program such as COPES. Therefore, a general programming practice is outlined here which in no way inhibits the use of the computer program in

its traditional role as an analytic tool, but allows for simple adaption to COPES. This approach is considered good programming practice and provides considerable flexibility of design options. Only five basic rules must be followed:

I. Write the code in subroutine form with the primary routine called as SUBROUTINE ANALIZ(ICALC). The name ANALIZ is compatable with the COPES program and ICALC is a calculation control. Note that subroutine ANALIZ may call numerous other subroutines as required to perform the necessary calculations.

II. Segment the program into INPUT, EXECUTION, and OUTPUT. The calculation control, ICALC, will determine the portion of the analysis code to be executed. ICALC=1; the program reads all data required to perform the analysis. Also, any initialization of constants which will be used repetively during execution is done here. This initial input information is printed here for later reference and for program debugging. ICALC=2; the program performs the execution phase of the analysis task. No data reading or printing is done here, except on user-defined scratch disc. Data may be printed here during program debugging, in which case it should be controlled by a print control parameter which is read during input. In this way, this print may be turned off after the program

is debugged, but may be used again during future program expansion debugging. The reason that printing is not allowed during execution is that when optimization is being done, the code will be called many times with ICALC=2, resulting in voluminous print. ICALC=3; the results of the anlaysis are printed. Also the essential input parameters which may have been changed during optimization should be printed here for easy reference. In summary, when:

ICALC = 1 Read input data.

ICALC = 2 Execute the analysis.

ICALC = 3 Print the results.

III. Store all parameters which may be design variables, objective functions or constraints in a single labeled common block called GLOBCM. The order in which they are stored is arbitrary. A listing of the COPES program should be checked to see how many parameters may be stored in GLOBCM (the dimension of ARRAY). Initial distribution of COPES allows for 1500 parameters.

IV. During execution or output, no parameters which are read during input should be updated. For example, if variable X is initialized during input, the execution segment must not update X such as X=X + 3.2. Instead a new variable, Y=X + 3.2 must be defined.

V. Write all programs in standard language, avoiding machine dependent capabilities such as seven letter FORTRAN names (CDC). While this guideline is not essential to the use of the analysis code within the COPES program, it makes the analysis code much more transportable between different computer systems, a capability which easily justifies a slight reduction in efficiency on a given machine.

Adherence to these guidelines not only leads to a more readable and machine independent computer code, but allows this code to be coupled to the COPES program without modification.

Having written the analysis code, it may be executed either with a simple main program or within the COPES program to perform the analysis. To insure that guideline IV is followed, the following main test program is recommended. Note that this program calls ANALIZ twice with ICALC=2 and ICALC=3, to show that the same result is obtained repetitively.

```
C   MAIN PROGRAM TO CHECK SUBROUTINE ANALIZ.
C   READ, EXECUTE, AND PRINT

    DO 10 ICALC=1,3
10  CALL ANALIZ (ICALC)
C   EXECUTE AND PRINT AGAIN TO BE SURE THE RESULTS
C   DO NOT CHANGE
    DO 20 ICALC=2,3
20  CALL ANALIZ (ICALC)
    STOP
    END
```

III. FINITE ELEMENT SOLUTION

A. REVIEW OF THE PREVIOUS ANALYSIS

Schafer [6] studied the one-dimensional model heat transfer solution and Corley [7] studied the two-dimensional model for an internally finned rotating heat pipe. Both used the same assumptions and boundary conditions based upon the analysis of Ballback [2], which are similar to those used in the Nusselt analysis of film condensation on a flat wall. The more important of those assumptions are:

1. steady state operation,
2. film condensation, as opposed to dropwise condensation,
3. laminar flow of the condensate film along both the fin and the trough,
4. static balance of forces within the condensate,
5. one-dimensional conduction heat transfer through the film thickness (no convective heat transfer in the condensate film),
6. no liquid - vapor interfacial shear forces,
7. no condensate subcooling,
8. zero heat flux boundary conditions on both sides of the wall section (symmetry conditions), as shown in Figure 5,
9. saturation temperature at the fin apex,

10. zero film thickness at the fin apex, and

11. negligible curvature of the condenser wall.

Purnomo [1] developed a two-dimensional Finite Element Method solution using a linear triangular finite element model as shown in Figure 4. Purnomo modified Corley's assumption that the fin apex was at the saturation temperature and allowed the value of the temperature at the apex to float. He assumed a parabolic temperature distribution along the fin surface.

Purnomo's statement of the problem for the formulation of the Finite Element Method as shown in Figure 5 is

$$\frac{\partial^2 T}{\partial x^2} + \frac{\partial^2 T}{\partial y^2} = 0 \tag{6}$$

with the boundary conditions:

a) along boundary 1, $-k \frac{\partial T}{\partial n} = h_1 (T - T_{sat})$

b) along boundary 3, $-k \frac{\partial T}{\partial n} = h_2 (T - T_\infty)$

c) along boundaries 2 and 4, $\frac{\partial T}{\partial n} = 0$

A detailed description of the numerical formulation is presented in his thesis.

Purnomo's computer program consisted of a main program and three subroutines;

a) the routine "COORD" used to define positions of system coordinate points,

b) the routine "FORMAF" used to formulate the Finite Element Method equations, and

c) the routine "BANDEC" as an equation solver for a symmetric matrix which has been transformed into banded form.

B. THE COMPUTER PROGRAM DEVELOPMENT

Purnomo's [1] two-dimensional finite element program is the basis for the present analysis code. The first task undertaken in the development of this thesis was to check Purnomo's code for validity. Theoretical results using Purnomo's code for the case of zero fins were compared to theoretical results and experimental data for a smooth tube obtained by Schafer [6]. Purnomo's results exceeded Schafer's results by a factor of approximately two. In studying this discrepancy, an error was discovered in Purnomo's code. In encoding formula (II-11) [1] for mass flow rate in FORTRAN, the ' sin ϕ ' term was dropped. The correct form of the equation is shown below.

$$\dot{M}_{tot}(x) = \frac{\rho^2 \omega^2 (R_0 + x\sin\phi)\delta^*(x)^2 \sin\phi}{3\mu_f}[\delta^*(x)\varepsilon + \delta^*(X)^2 \tan\alpha] \tag{7}$$

The effect of this error was that the film thickness along the length of the condenser remained very small. Purnomo's original code consequently indicated a much higher value of heat transfer rate than the correct analysis did. In a correct analysis code, the condensate film thickness grows continuously until the condensate reaches the evaporator. A plot of the condensate film thickness calculations from Purnomo's original code for the case of a smooth tube, his corrected code, and Schafer's program for a smooth tube are shown in Figure 6. Purnomo's corrected code for the case of zero fins agreed to within 8 percent of the results for heat transfer rate using Schafer's [6] analysis for a smooth tube.

Since no experimental data existed for further comparison of the finned model, the next task undertaken was to adapt the analysis code to permit automated design and sensitivity analysis using COPES/CONMIN. Many modifications were made, some of which are mentioned here. The program was rewritten in subroutine form and segmented into input, execution, and output sections to make it compatible with COPES. Since COPES was written to use single precision mathematics and the analysis code uses double precision to allow for possible ill conditioning, the subroutine ANALIZ also makes the transformation from single to double precision. The initial value of the film thickness is calculated within the code based on

a formulation by Sparrow and Gregg [13]. Previously a constant value based on the same analysis was used.

The modified code can be used alone for analysis of a given geometry or it can be used in conjunction with COPES/CONMIN for a single analysis, sensitivity analysis, or automated design (optimization).

A listing of the revised computer program is included as Appendix B.

C. SENSITIVITY ANALYSIS

The sensitivity analysis feature of COPES/CONMIN was used to obtain data for the heat transfer rate Q as a function of various design variables. All design variables were held constant except the one of interest which was varied as specified in block Q of the COPES input data.

For example, to obtain the heat transfer rate Q for 26 different values of rotational speed, the following data input is required:

```
$ Block P
  1
  15
$ Block Q
  16, 26
  3000., 100., 200., 300., 400., 500., 600., 700.,
  800., 900., 1000., 2000., 3000., 4000., 5000., 6000.,
  7000., 8000., 9000., 10000., 11000., 12000., 13000.,
  14000., 15000., 3000.
```

For each plot only the variable on the ordinate is changed. All other design variables remain constant. The basic design used for sensitivity plots has the design variable values shown below:

TABLE II

BASIC DESIGN FOR SENSITIVITY ANALYSIS

condenser length	CLI	= 9.0 inches
cone half angle	CANGL	= 1.0 DEGREES
condenser radius	RBASEI	= 0.775 inches
wall thickness	THICKI	= 0.03125 inches
fin height	B	= 0.025 inches
speed of rotation	RPM	= 3000 RPM
saturation temperature	TSAT	= 100° F
ambient temperature	TINF	= 60° F
outside heat transfer coefficient	HINF	= 5000 Btu/HR FT^2 °F
fin half angle	FANGL	= 10
ratio of trough width to fin base width	ZOA	= 12.8
number of fins	ZFIN	= 40.0

Note: In the code, ZFIN appears on the left hand side of an equation and is therefore by strict definition not a "design variable". ZFIN is calculated from the values of CBASE, EZERO, and EPSO.

D. DESIGN OPTIMIZATION

Counting parameters such as external heat transfer coefficient HINF, there are thirteen possible design

variables. Nine of these are geometric or functional parameters such as wall thickness, fin height, and speed of rotation. The design variables, possible constraint functions, and the objective function appear in the Global common block GLOB1 in the code and are listed below by fortran name for clarity.

DESIGN VARIABLES

CLI
CANGL
RBASEI
R2I
THICKI
BFIN
TZ
TSS
TINF
HINF
FANGL
ZOA
RPM

CONSTRAINT FUNCTIONS

ZFIN
BOA
DIFF

OBJECTIVE FUNCTION

QTOT

There are a wide variety of design problems that can be pursued with the code. For example one might wish to

determine the smallest length condenser section and the best internal geometry for a specified heat transfer rate--perhaps to cool an electric motor.

To maximize heat transfer rate through the condenser wall the designer adds a number of fins to increase the inside surface area. As more and more fins are added however, the cross-sectional area for conduction through each fin is decreased. Also, the film thickness of the condensate in the though increases and in fact could completely cover the fins and substantially reduce heat transfer through the fin. So there should exist some optimum combination of number of fins, fin height, fin half angle, and ratio of trough width to fin width that will permit maximum heat transfer rate.

The design study undertaken was to determine the fin height, fin half angle, and fin spacing which would yield the maximum heat transfer possible. The design variables then were BFIN, FANGL, and ZOA. Other potential design variables were held constant. The objective function to be maximized was QTOT, heat transfer rate out of the condenser.

For comparison, the theoretical upper limit on heat transfer was calculated based on an external surface temperature equal to the working fluid saturation temperature. This assumes that there is no thermal resistance across the condensate and the condenser wall. When this upper limit was used, the maximum heat transfer rate was predicted to be

63,322 BTU/HR using the following formula:

$$Q_{max} = h\ 2\pi\bar{r}\ L\ (Twall - T\infty) \qquad (8)$$

where

h = outside convective heat transfer coefficient ($5000\ BTU/HR \cdot FT^2 \cdot {}^\circ F$)

$\bar{r}$ = average outside radius of condenser wall (0.0056 ft)

L = condenser length (0.75 feet)

$Twall$ = temperature of the outside wall (100°F)

$T\infty$ = ambient temperature (60°F)

Certain constraints were placed on the design based on engineering judgement. For example, the number of fins was not allowed to exceed 400 and the minimum fin half angle allowed was 10 degrees. These values were based on structural and manufacturing considerations.

IV. RESULTS

A. INTRODUCTION

The computer code was used in conjunction with the COPES/ CONMIN program for sensitivity analysis and design optimization for maximum heat transfer rate. Numerical results are discussed below.

B. SENSITIVITY ANALYSIS

As shown in Figures 7 and 8, the heat transfer rate increases for an independent increase of the design variable of interest, but levels off at the theoretical maximum heat transfer rate for the given overall geometry. The heat transfer rate then is limited by the external resistance which has become the controlling factor.

In Figure 9, the heat transfer rate increases in a similar way for an increase in external heat transfer coefficient. Again, the rate of increase appears to lessen for external heat transfer coefficients above 10,000 $BTU/HR \cdot FT^2 \cdot {}^\circ F$ due to other limiting resistances. In Figure 10 the heat transfer rate is observed to increase linearly with condenser length. This is expected since Q is a function of the area and for small values of ϕ the area varies directly with length. In Figure 11, Q is seen to rise in a non-linear manner for an increase in

cone half angle, ϕ. This is because the internal heat transfer coefficient, h_i is a non-linear function of cone half angle.

Purnomo [1] concluded that the heat transfer rate continuously increased as the fin half angle decreased and that this was largely a result of the fact that the number of fins increased at the same time. He also stated that the increase in heat transfer was only slight when the fin half angle was less than 11 degrees. This is somewhat misleading in that in his analysis the number of fins was being changed with every change in fin half angle. This author has drawn a different conclusion. Figure 12 shows a plot of heat transfer rate vs. varying fin half angle all for a condenser with 40 fins. The heat transfer rate as a function of fin half angle rises sharply from 1-11 degrees and continues to rise, but less steeply, as the fin half angle increases beyond 11 degrees. The maximum fin half angle possible with 40 fins is 67.5 degrees.

C. CONSTRAINED OPTIMIZATION

In the design problem undertaken to determine the optimum internal geometry for maximum heat transfer, numerous runs were made for condensers made of copper, stainless steel, and a ceramic material. These materials have thermal conductivity values of 231,9, and 1.0 BTU/HR·FT·°F respectively. It was expected that for each material a different optimum

design would emerge. The results, therefore, were unexpected. From different starting points (original designs), some outside the feasible region, and for external heat transfer coefficients from 1000-50,000 BTU/HR·FT^2·°F, the same optimum design for maximum heat transfer was reached, which is outlined in Table III below. Each material, of course, had a different heat transfer rate even though the geometry for maximum heat transfer was the same.

TABLE III

CONSTRAINED OPTIMIZATION RESULTS FOR COPPER STAINLESS STEEL, AND CERAMIC MATERIAL

Optimum Design for All Materials

fin height	0.023 inches
fin half angle	10.0 degrees
number of fins	400.0
ratio of trough width to fin base width	0.5

MATERIAL	$h_{external}$ (BTU/HR·FT^2·°F)	HEAT TRANSFER RATE (BTU/HR)
Copper	1000	13,822
	50,000	296,560
Stainless Steel	1000	10,600
	50,000	38,650
Ceramic Material	1000	3844
	50,000	5237

D. UNCONSTRAINED OPTIMIZATION

The results of the constrained optimization runs were so unexpected that it was decided to remove the constraint on the number of fins and the trough width to fin width ratio and repeat the optimization runs. If another identical design was again reached for all materials, the code would have to be considered in error. The results are presented in Table IV.

The optimum design for maximum heat transfer was different for each material and each heat transfer coefficient. For each material, the optimum fin height b increases as the external heat transfer coefficient increases. This provides more fin surface area to increase heat transfer rate as the outside heat transfer resistance no longer controls. Despite a fifty-fold change in heat transfer coefficient, each material maintained essentially a constant δ^*/b ratio. The higher conductivity materials required less exposed fin surface than the lower conductivity materials. Copper, for instance, could be 57 percent covered by the trough condensate while the ceramic material was only 14-19 percent covered for maximum heat transfer.

In each material the number of fins decreased for an increase in external heat transfer coefficient. This provides more space in the troughs to carry the increased condensate which results. Otherwise, fin performance would be degraded.

TABLE IV

UNCONSTRAINED OPTIMIZATION RESULTS
FOR COPPER, STAINLESS STEEL, AND CERAMIC MATERIAL

Material	$h_{external}$ ($BTU/HR \cdot FT^2 \cdot {}^\circ F$)	δ^* (Inches)	b (Inches)	c/a	Number of fins	Q (BTU/HR)	δ^*/b
Copper	1000	0.004	0.007	0.0001	2040	13823	0.57
Copper	50000	0.011	0.019	0.0001	723	298330	0.58
Stainless Steel	1000	0.004	0.013	0.021	1046	10602	0.31
Stainless Steel	50000	0.005	0.016	0.234	684	37138	0.31
Ceramic	1000	0.002	0.014	0.001	953	3843	0.14
Ceramic	50000	0.003	0.016	0.001	862	5237	0.19

NOTE: Geometrical parameters used in the above table are defined in Figures 4 and 5.

For an increase in external heat transfer coefficient the highest percentage change in heat transfer rate occurs in the material with the highest thermal conductivity. In the ceramic material, for instance, a fifty-fold increase in external heat transfer coefficient results in only a 36 percent increase in heat transfer rate. In the copper condenser, the heat transfer rate is increased by a factor of 21. This shows the strong dominance of the wall resistance to heat transfer in the ceramic material.

For all three materials, the trough is essentially eliminated for the optimum design. Why the trough is not entirely eliminated in the stainless steel condenser section is not clear.

Because the manufacture of a very large number of fins in a small diameter condenser is not practical, it is desirable to use the constrained optimization results as a design guideline.

A comparison of the unconstrained and constrained design for a copper condenser section, for instance, shows that using the more realistic number of 400 fins and its proper fin height b instead of 2040 fins results in a heat transfer rate difference of only 0.6 percent. This is shown clearly in Table V.

TABLE V

Comparison of Unconstrained and Constrained Optimum Design for a Copper Condenser Section with External Heat Transfer Coefficient of 50000 $BTU/HR \cdot FT^2 \cdot {}^\circ F$.

	b (inches)	Number of Fins	Q (BTU/HR)	%Difference in Q
constrained	0.023	400	296,560	----
unconstrained	0.019	2040	298,330	0.6%

The author's conclusion is that within the realm of what can now be manufactured, the same basic design is best for all materials regardless of the external heat transfer coefficient. That is, the designer should machine as many fins as the condenser material and the manufacturing process will allow. The code should be used to determine the fin height to avoid degrading the fin efficiency by too high a level of condensate in the trough. As seen in Table V, it may be possible to lower the constraint on the number of fins more than once and compare the resulting heat transfer rate to achieve an effective but less expensive design to manufacture.

E. A CAUTIONARY NOTE

The reader should be cautioned that when an analysis code based on assumptions made for certain conditions is linked to an optimizer, the code may change the geometry or other conditions such that the original assumptions are no longer

valid. Judicious use of side constraints and engineering common sense can prevent disaster. Numerical solutions must not be blindly accepted.

It should be noted that all calculations in this thesis were made with water as the working fluid. Sensitivity data were run with an original rotational speed of 3000 RPM while design optimization was done for a rotational speed of 3600 RPM. Different combinations of operating parameters, working fluids, and condenser section materials may not behave in a predictable manner. While certain trends are shown for the particular condenser section studied here, it would be preferable to modify the code to analyze the particular problem at hand than to extrapolate these findings to too broad an application.

While a design improvement is almost certain using an optimization routine like COPES/CONMIN, there is no guarantee that a global optimum has been found. Therefore, for a given design problem, the designer should start the optimization process from several different original designs and compare the results.

V. CONCLUSIONS

1. The computer code is valid and can be used for single analysis, sensitivity analysis, and automated design of an internally finned rotating heat pipe. It can also be used to generate numerical data to find a correlation useful for design.

2. For zero fins, the code converges to the results for a smooth tube obtained by Schafer [6].

3. The heat transfer rate increases for an independent increase in fin half angle, rotational speed, and number of fins but levels off at the theoretical maximum heat transfer rate for the heat pipe. The heat transfer rate continues to increase for an increase in condenser length and cone half angle.

4. Within the realm of known materials, maximum heat transfer occurs for the same fin geometry regardless of the external heat transfer coefficient. That is, for a given condenser radius, one should machine as many fins as possible with the condenser material used to maximize heat transfer. The code should be used to determine the correct fin height to avoid degradation of fin performance by too high a level of condensate in the trough.

5. When the constraint on minimum fin half angle is removed the optimum design is a large number of very thin fins. A practical design then might be a series of long rectangular fins.

VI. RECOMMENDATIONS

Based upon the calculated results of this thesis, the following recommendations are made.

1. Build an internally finned heat pipe to obtain experimental data for comparison to the analytical predictions.
2. Generate a new model to analyze the case where cone half angle is zero. This would greatly decrease the cost of manufacture, and permit the use of internally finned heat exchanger tubing which is commercially available.
3. Analyze different shaped fins, including rectangular.
4. Obtain numerical data using the code to attempt to find a correlation for the heat transfer rate as a function of important variables:

   Q = Q (temperature, condenser geometry, fluid properties, condenser material properties, heat transfer coefficients, etc.)
5. Modify the code to include as constraint functions structural failure modes such as rupture due to the internal spinning mass, buckling, and whirling.

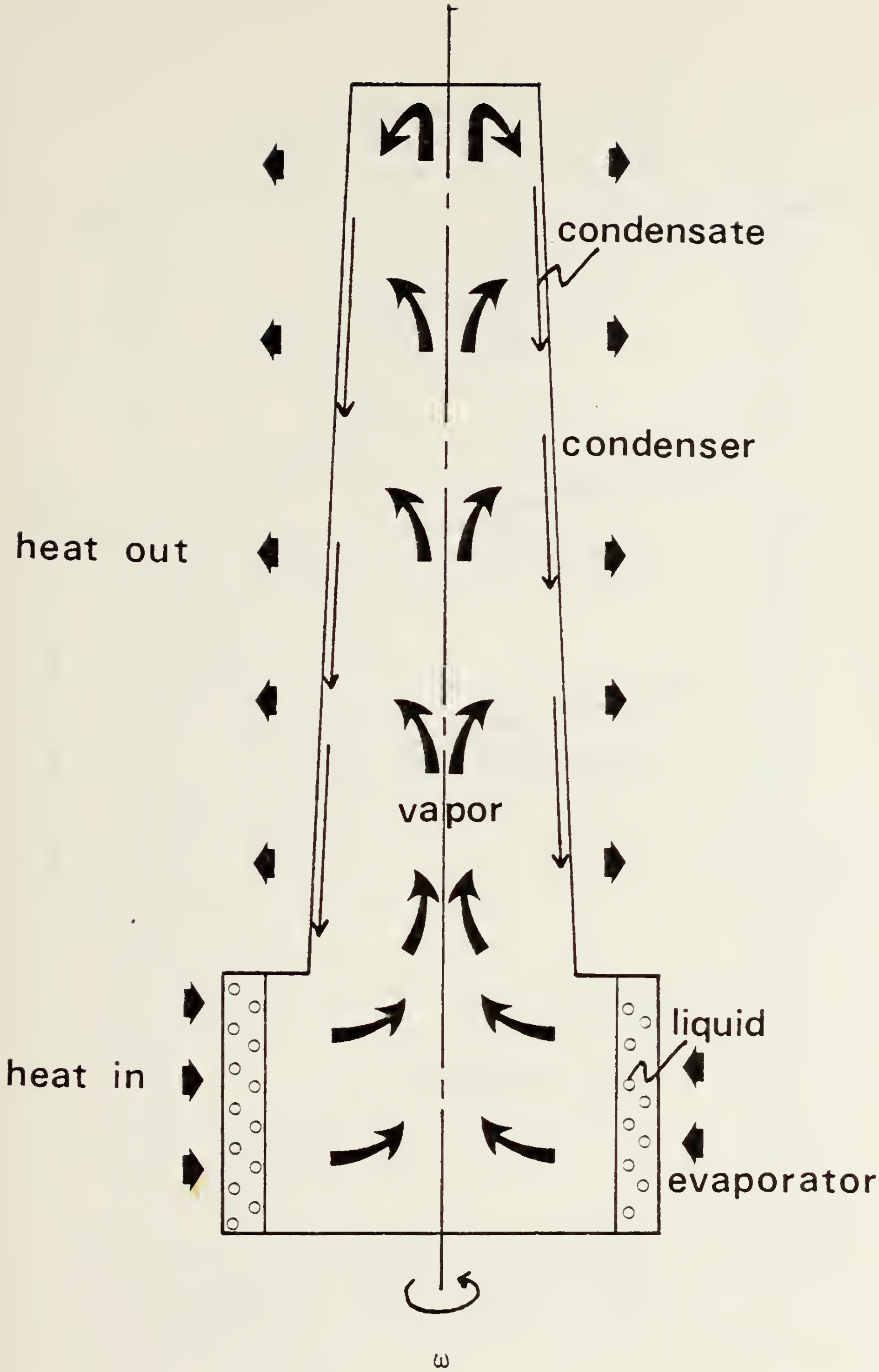


Figure 1. Schematic Drawing of a Rotating Heat Pipe

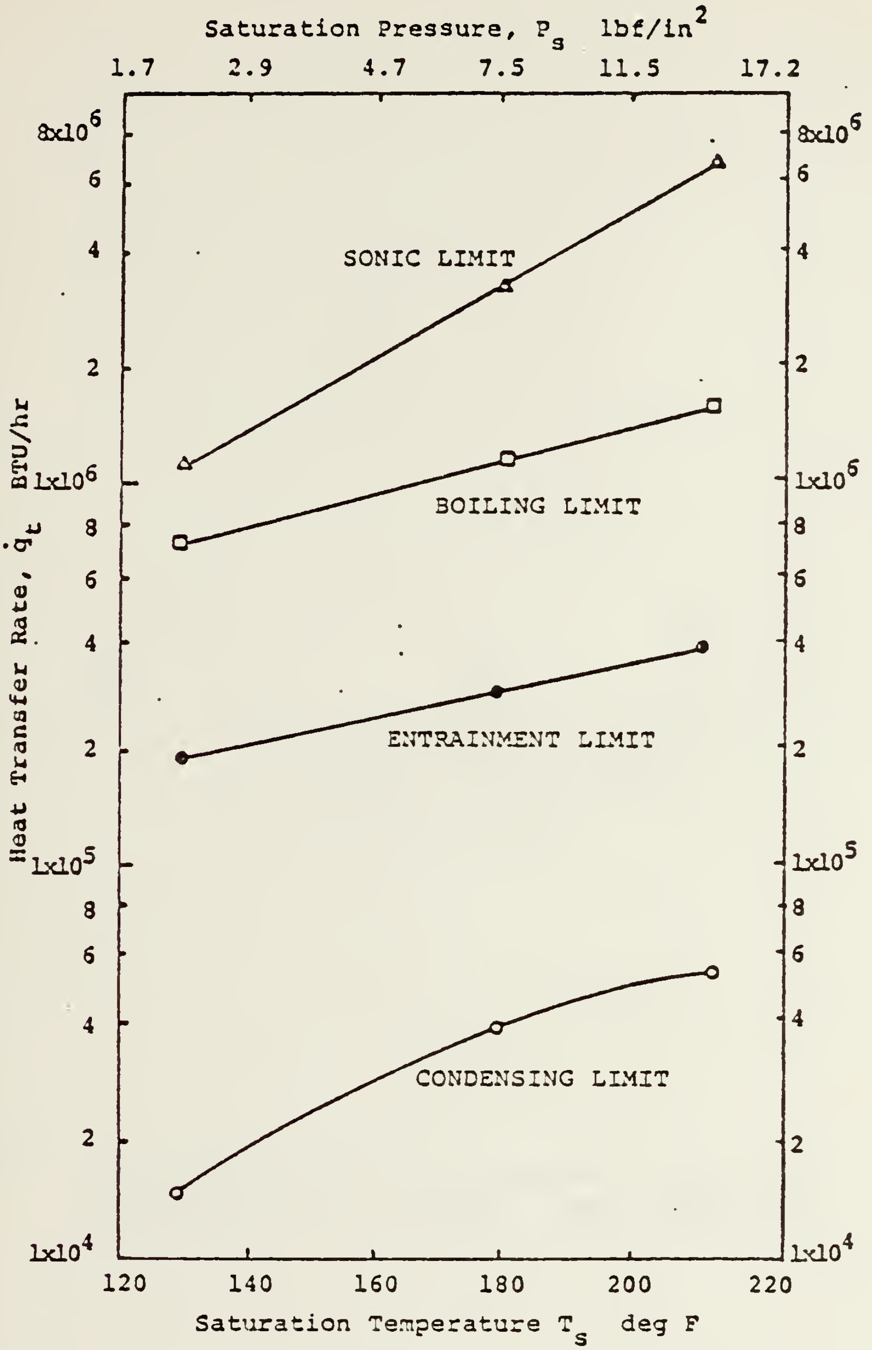


Figure 2. Operating Limits of a Typical, Water-Filled Rotating Heat Pipe

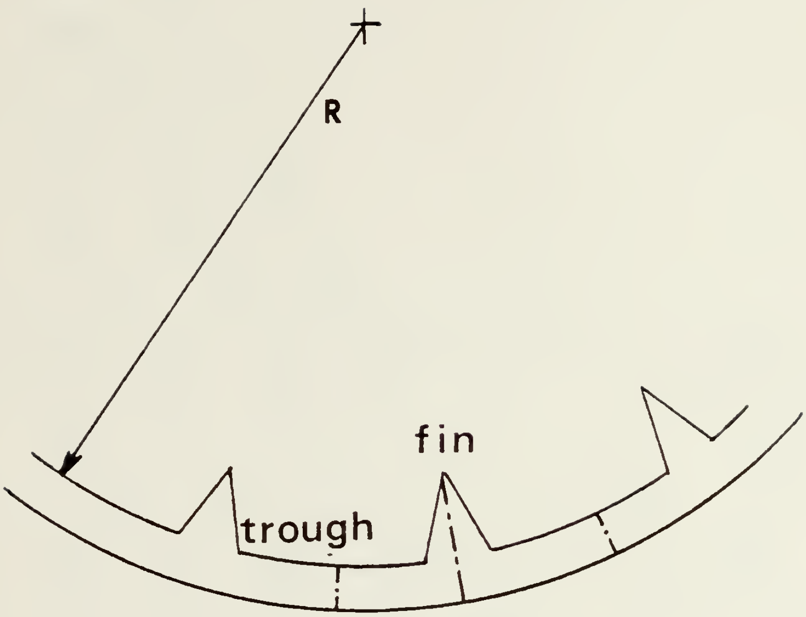
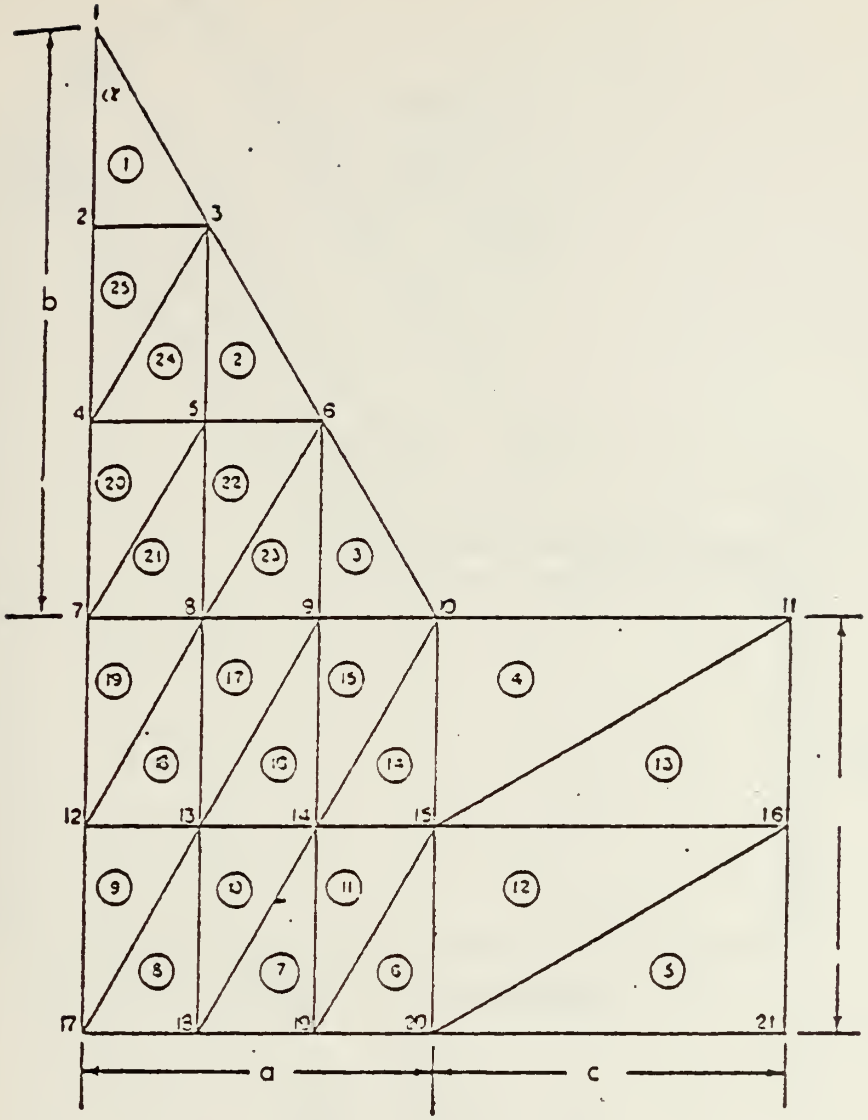


Figure 3. Internally Finned Condenser Geometry, Showing Fins, Troughs, and Lines of Symmetry



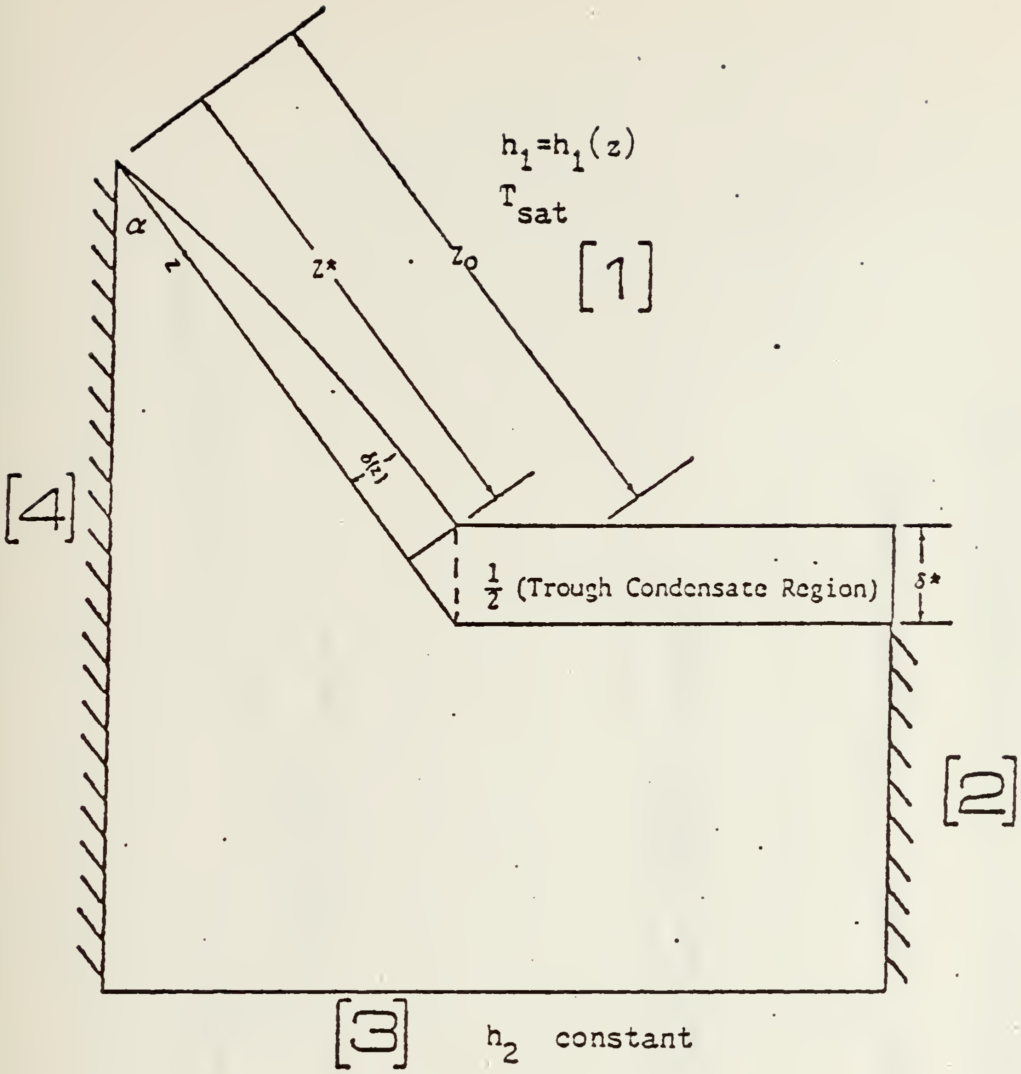
t = 0.03125 in

b = 0.025 in

R_2 = 0.762 in

ϕ = 1°

Figure 4. Condenser Geometry Considered with 25 Linear Triangular Finite Elements



$$\frac{\partial^2 T}{\partial x^2} + \frac{\partial^2 T}{\partial y^2} = 0$$

b.c.

a) $-k\frac{\partial T}{\partial n} = h_1 (T - T_{sat})$ Along Boundary [1]

b) $-k\frac{\partial T}{\partial n} = h_2 (T - T_\infty)$ Along Boundary [3]

c) $\frac{\partial T}{\partial n} = 0$ Along Boundaries [2] and [4]

Figure 5. Differential Equation and Boundary Conditions Considered in the Analysis of Purnomo [1]

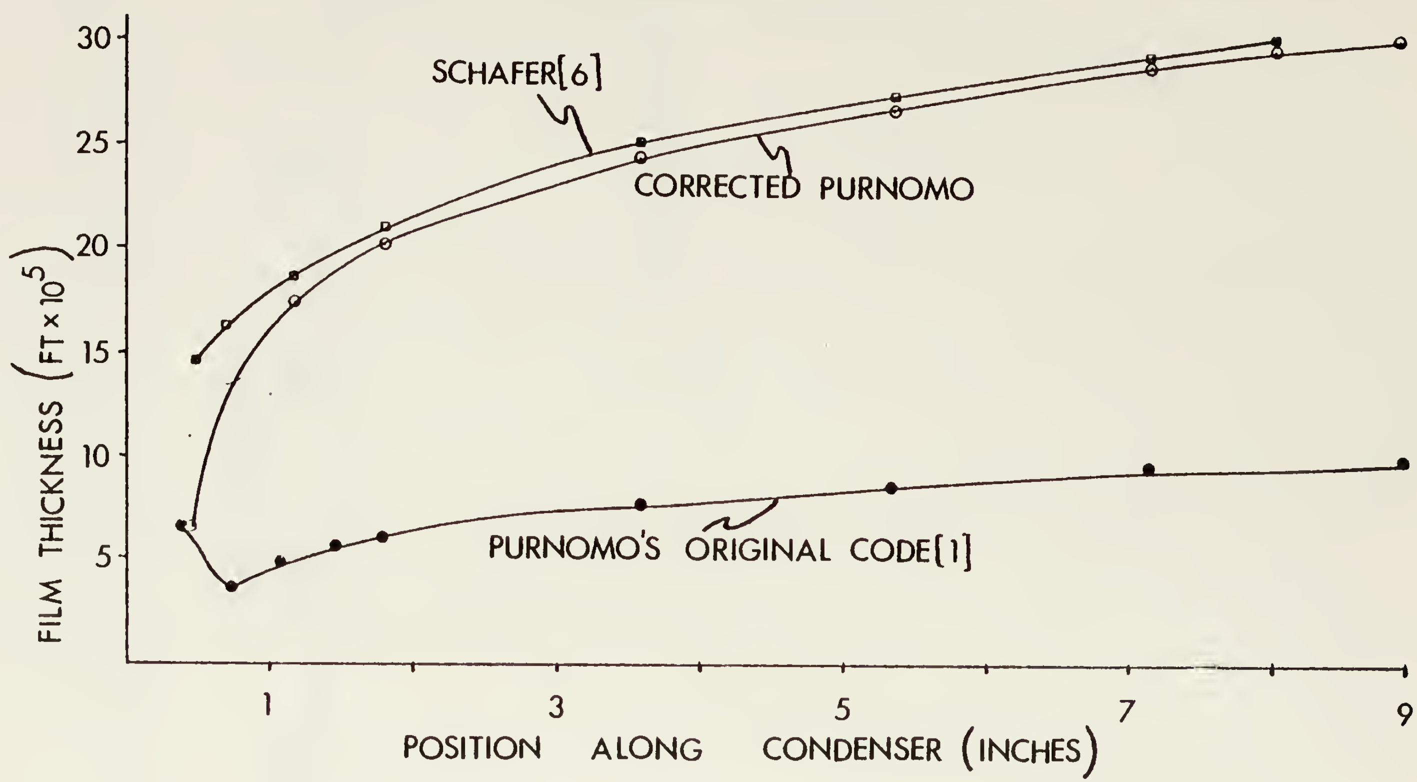


Figure 6. Comparison of Trough Condensate Film Thickness vs. Length Along the Condenser

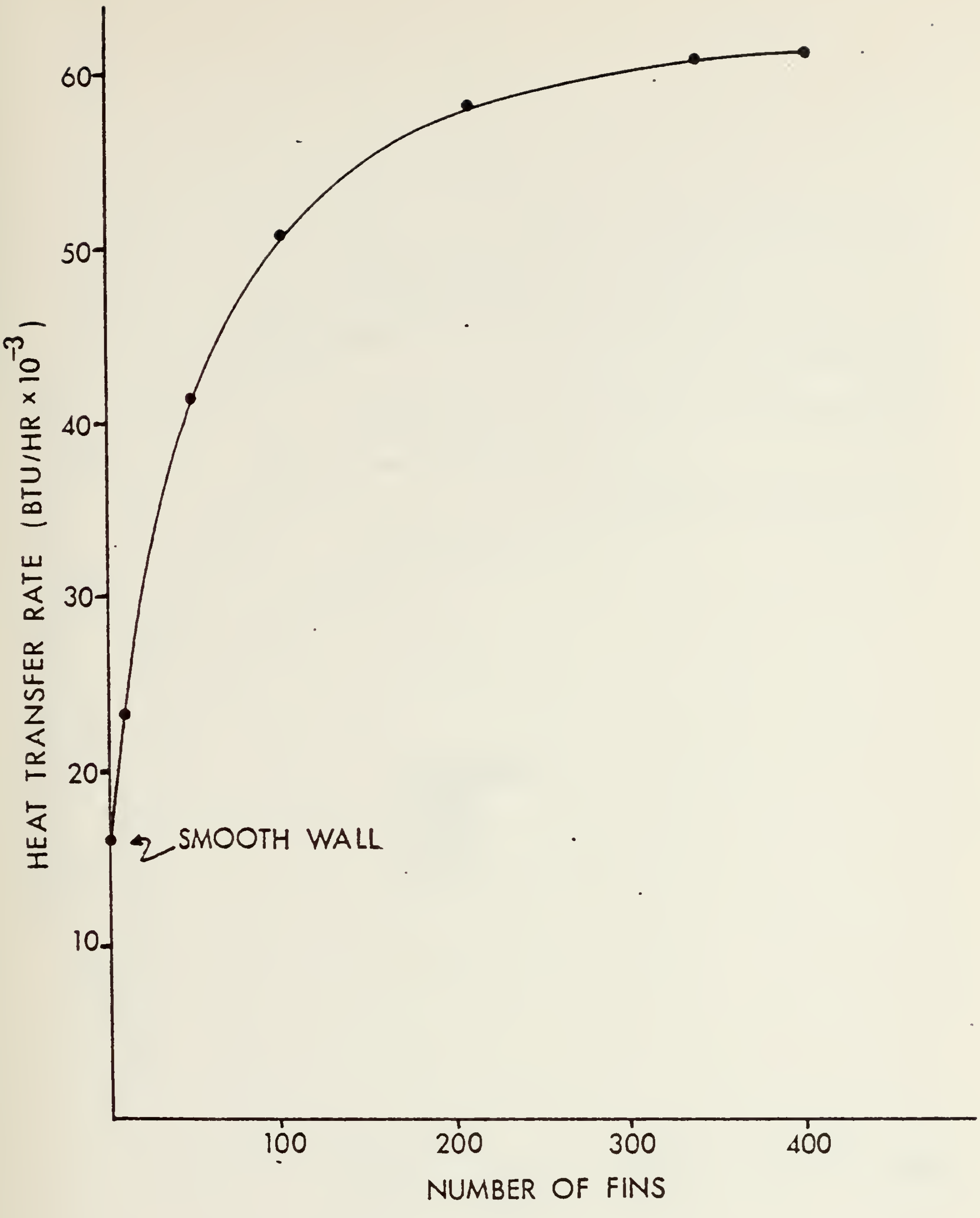


Figure 7. Heat Transfer Rate (Q) vs. Number of Fins

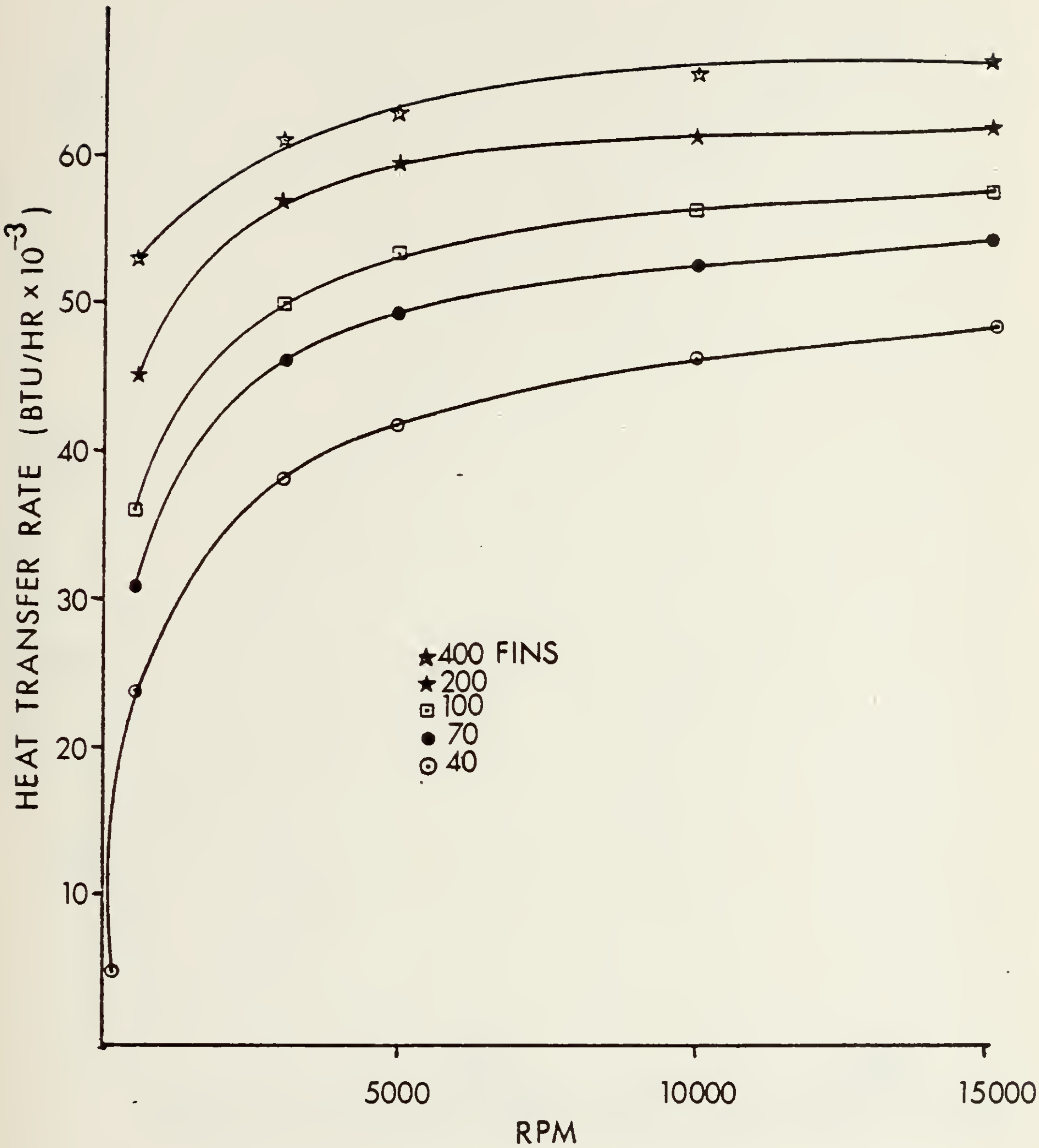


Figure 8. Heat Transfer Rate (Q) vs. RPM

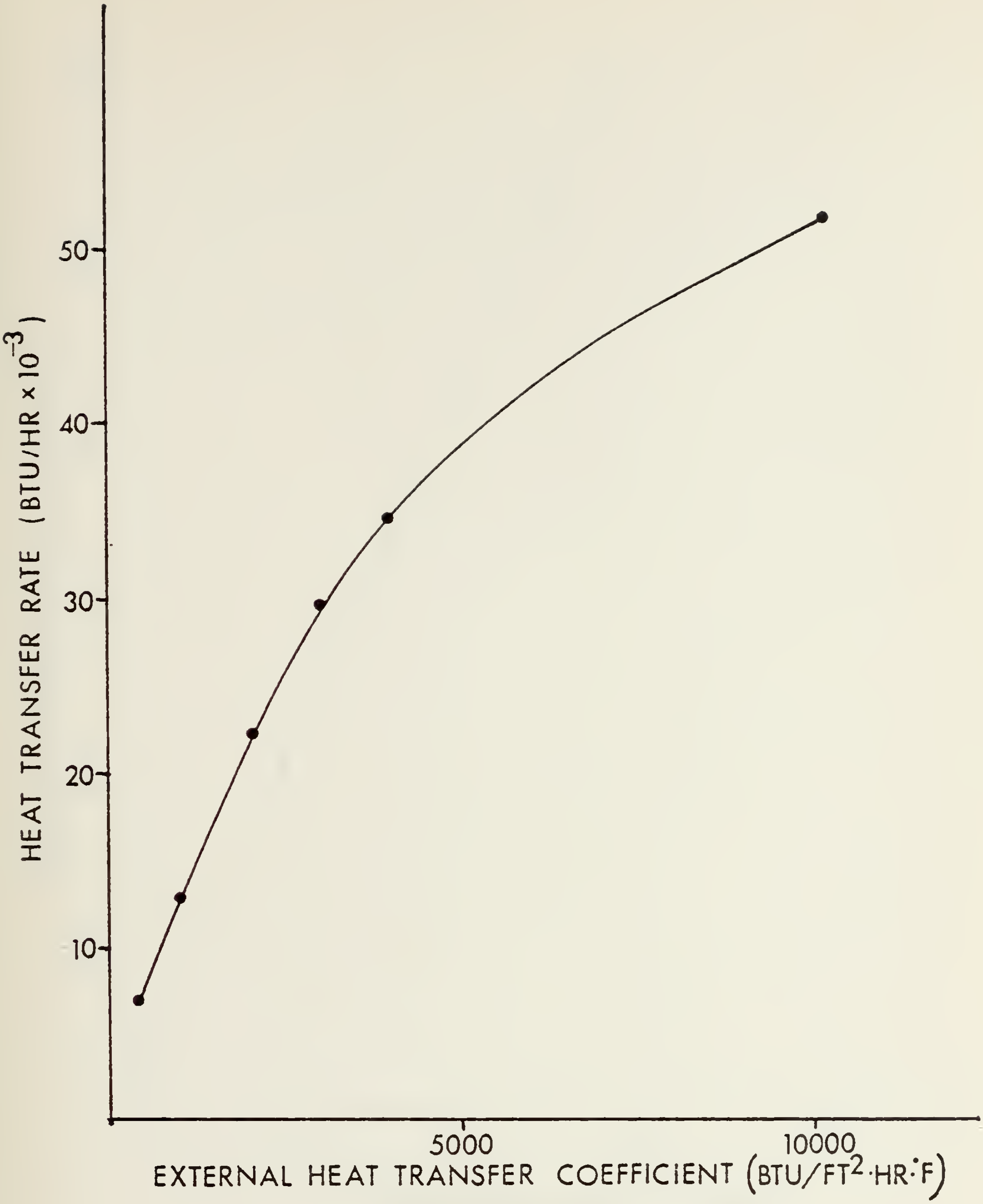


Figure 9. Heat Transfer Rate (Q) vs. External Heat Transfer Coefficient

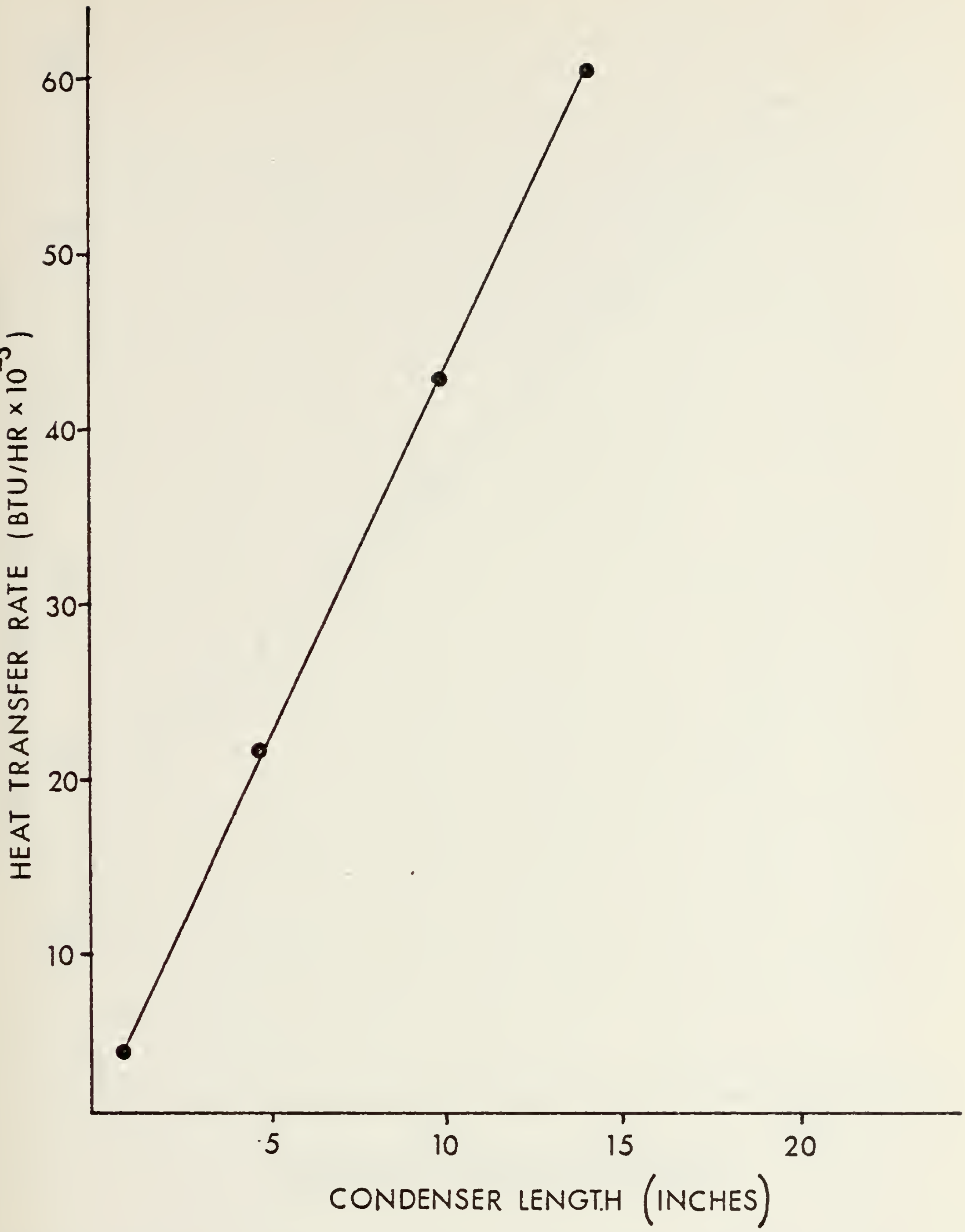


Figure 10. Heat Transfer Rate (Q) vs. Condenser Length

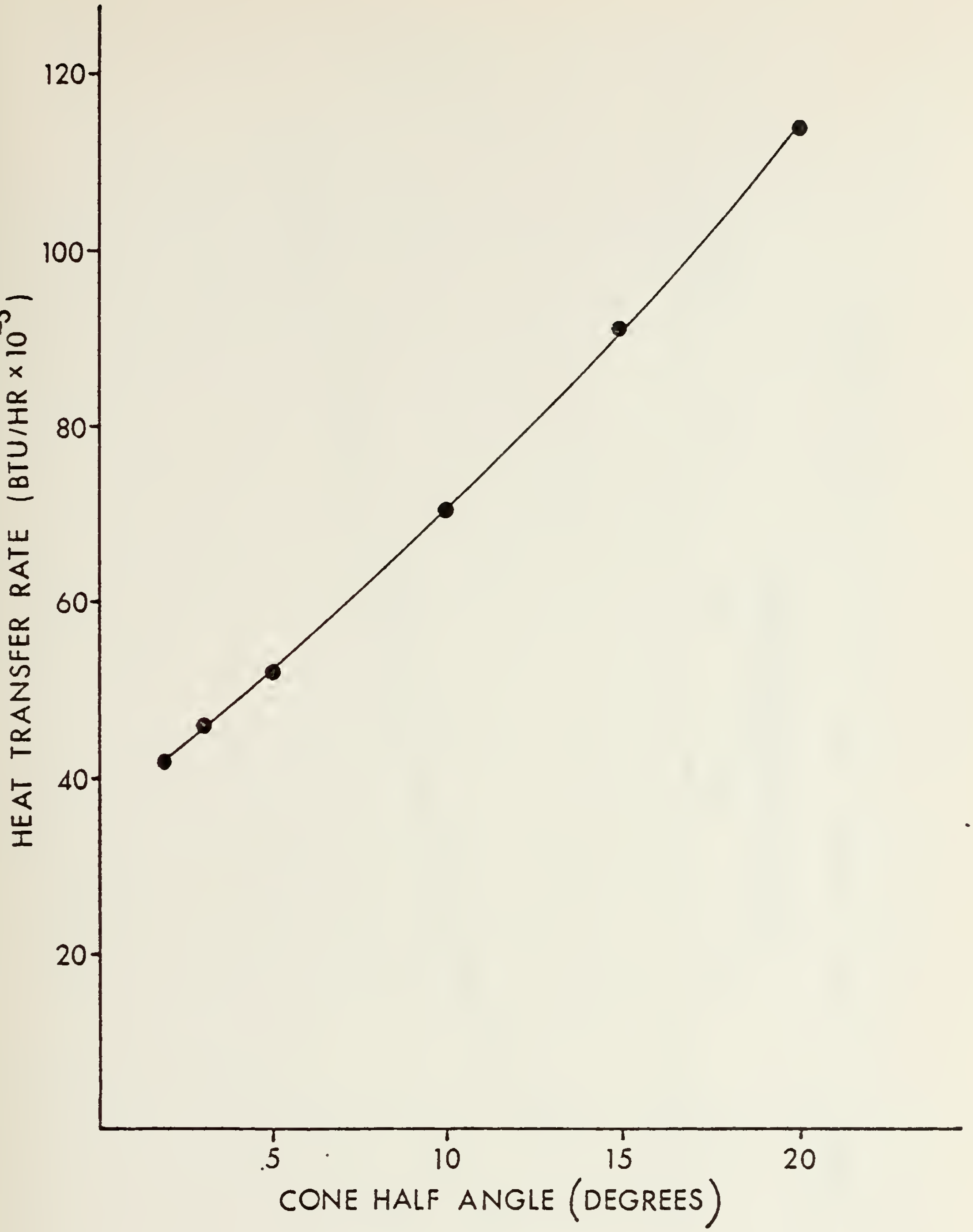


Figure 11. Heat Transfer Rate (Q) vs. Cone Half Angle

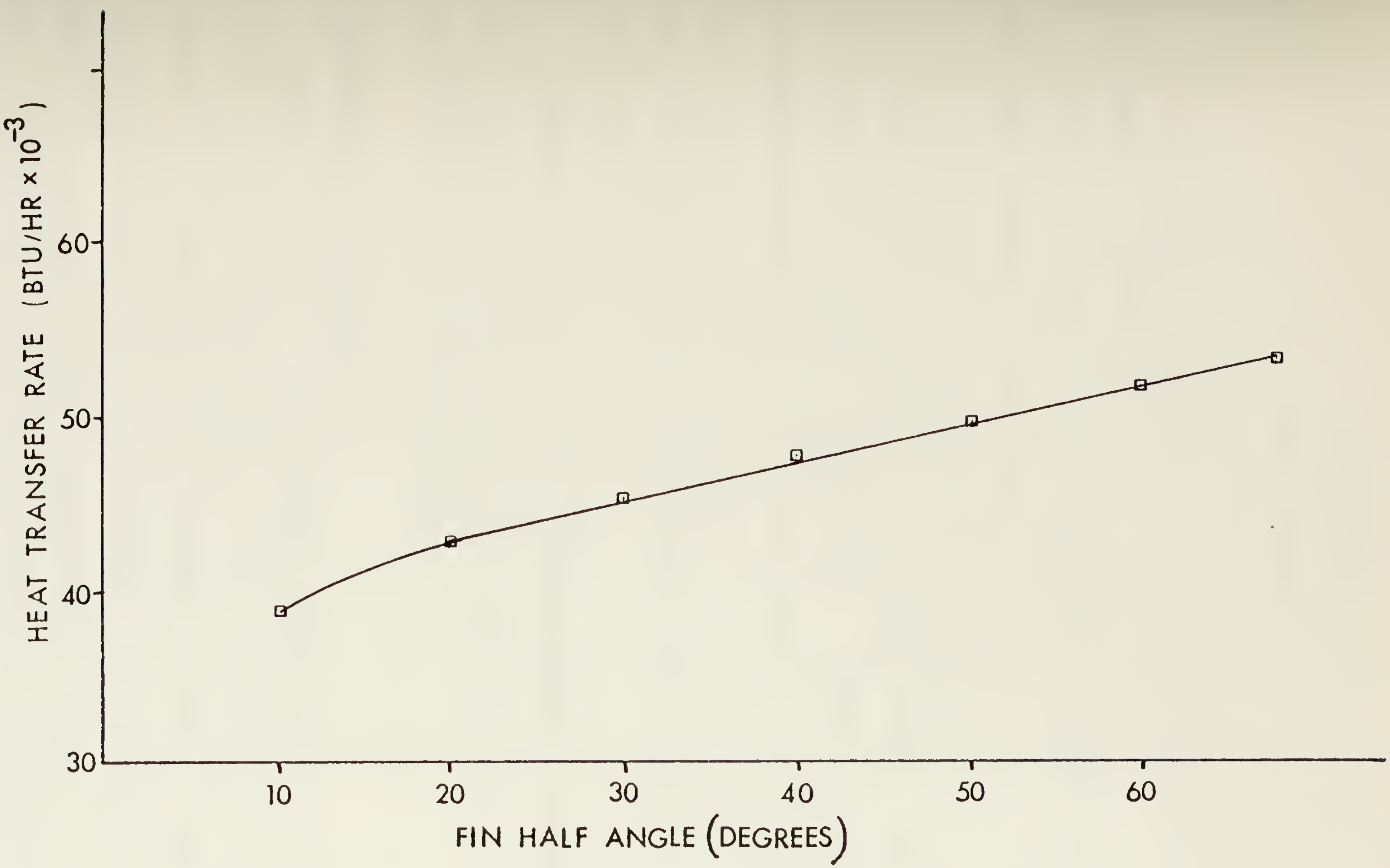


Figure 12. Heat Transfer Rate (Q) vs. Fin Half Angle

APPENDIX A

USER'S MANUAL

This appendix describes the data cards for the use of the computer program.

The data is divided into the COPES/CONMIN section and the heat pipe analysis program section.

The COPES data is segmented into "blocks" for convenience. All formats are alphanumeric for TITLE and END cards, F10 for real data, and I10 for integer data. Comment cards may be inserted anywhere in the data deck prior to the END card and are identified by a dollar sign ($) in column one. The COPES data deck must terminate with an end card containing the word 'END' in columns 1-3.

Information is included in this appendix pertaining to data needed for single analysis, sensitivity analysis, or optimization using COPES/CONMIN.

UNFORMATTED DATA INPUT

While the user's sheet defines COPES data in formatted fields of ten, the data may actually be read in a simplified fashion by separating data by commas or one or more blanks. If more than one number is contained on an unformatted data card, a comma must appear somewhere on the card. If exponential numbers such as 2.5+10 are read on an unformatted card, there must be no embedded blanks. Unformatted cards may be intermingled with formatted cards. Real numbers on an

unformatted card must have a decimal point.

EXAMPLES

Unformatted data;

```
5,7,1.3,1.0+20,-5.1

5,7,1.3,1.0+20,,-5.1

5 7 1.3 1.0+20,, -5.1

5      7  1.3, 1.0+20 0 -5.1
```

Equivalent formatted data;

col→ 10	20	30	40	50	60	70	80
5	7	1.3	1.0+20	0	-5.1		

DATA BLOCK A

DESCRIPTION: COPES TITLE CARD

FORMAT: 20A4

1	2	3	4	5	6	7	8
Title							

FIELD	CONTENTS
1-8	Any 80 Character title may be given on this card.

DATA BLOCK B

DESCRIPTION: PROGRAM CONTROL PARAMETERS

FORMAT: 7I10

1	2	3	4	5	6	7	8
NCALC	NDV	NSV					

FIELD	CONTENTS
1	NCALC: Calculation control 0- Read input and stop. Data of blocks A,B and V is required. Remaining data is optional. 1- One cycle through program. The same as executing ANALIZ stand-alone. Data of blocks A,B and V is required, remaining data is optional. 2- Optimization. Data of Blocks of A-I and V is required. Remaining data is optional. 3- Sensitivity analysis. Data of blocks, A,B,P,Q and V is required. Remaining data is optional.
2	NDV: Number of independent design variables in optimization.
3	NSV: Number of variables on which sensitivity analysis will be performed.

DATA BLOCK C Omit if NDV=0 in block B

DESCRIPTION: INTEGER OPTIMIZATION CONTROL PARAMETERS

FORMAT: 7I10

1	2	3	4	5	6	7	8
IPRINT	ITMAX	ICNDIR	NSCAL	ITRM	LINOBJ	NACMXI	

FIELD	CONTENTS
1	IPRINT- Print control used in the optimization program CONMIN. 0- No print during optimization. 1- Print initial and final optimization information. 2- Print above plus objective function value and design variable values at each iteration. 3- Print above plus constraint values, direction vector and move parameter at each iteration. 4- Print above plus gradient information. 5- Print above plus each proposed design vector, objective function and constraint values during the one dimensional search.
2	ITMAX: Maximum number of optimization inter-actions allowed. Default=20.
3	ICNDIR: Conjugate direction restart parameter. GT. 0- Scale design variables to order of magnitude one every NSCAL iterations. LT. 0- Scale design variables according to user in-put scaling values. If not zero, NDV + 1 is recommended.
5	ITRM: Number of consecutive iterations which must satisfy relative or absolute convergence criterion before optimization process is terminated. Default=3.
6	LINOBJ: Linear objective function identifier. If the optimization objective is known to be a linear function of the design variables, set LINOBJ=1. Default=Non-linear.

7 NACMX1: One plus the maximum number of active constraints anticipated. Default-NDV=2.

DATA BLOCK D Omit if NDV=0 in block B

DESCRIPTION: FLOATING POINT OPTIMIZATION PROGRAM PARAMETERS

FORMAT: 7F10

1	2	3	4	5	6	7	8
FDCH	FDCHM	CT	CTMIN	CTL	CTLMIN	THETA	

NOTE: Two cards are read here.

FIELD	CONTENTS
1	FDCH: Relative change in design variables in calculating finite difference gradients. Default=0.01.
2	FDCHM: Minimum absolute step in finite difference gradient calculations. Default=0.001.
3	CT: Constraint thickness parameter. Default==-0.05.
4	CTMIN: Minimum absolute value of CT considered in the optimization process. Default=0.004.
5	CTL: Constraint thickness parameter for linear constraints. Default=-0.01.
6	CTLMIN: Minimum absolute value of CTL considered in the optimization process. Default=0.001.
7	THETA: Mean value of push-off factor in the method of feasible directions. Default=1.0.

DATA BLOCK D   Omit if NDV=0 in block B

FORMAT: 4F10

1	2	3	4	5	6	7	8
DELFUN	DABFUN	ALPHAX	ABOBJ1				

FIELD	CONTENTS
1	DELFUN: Minimum relative change in objective function to indicate convergence of the optimization process. DEFAULT=0.001.
2	DABFUN: Minimum absolute change in objective function to indicate convergence of the optimization process. DEFAULT=0.001 times the initial objective value.
3	ALPHAX: Maximum fractional change in any design variable for first estimate of the step in the one-dimensional search. DEFAULT=0.1.
4	ABOBJ1: Expected fractional change in the objective function for first estimate of the step in the one-dimensional search. DEFAULT=0.1.

REMARKS

1) The DEFAULT values for these parameters usually work well.

DATA BLOCK E   Omit if NDV=0 in block B

DESCRIPTION: TOTAL NUMBER OF DESIGN VARIABLES, DESIGN OBJECTIVE IDENTIFICATION AND SIGN

FORMAT: 2I10,F10

1	2	3	4	5	6	7	8
NDVTOT	IOBJ	SGNOPT					

FIELD	CONTENTS
1	NDVTOT: Total number of variables linked to the design variables. This option allows two or more parameters to be assigned to a single design variable. The value of each parameter is the value of the design variable times a multiplier, which may be different for each parameter. DEFAULT=NDV.
2	IOBJ: Global variable location associated with the objective function in optimization
3	SGNOPT: Sign used to identify whether function is to be maximized or minimized. +1.0 indicates maximization. -1.0 indicates minimization. If SGNOPT is not unity in magnitude, it acts as a multiplier as well, to scale the magnitude of the objective.

DATA BLOCK F   Omit if NDV=0 in block B

DESCRIPTION: DESIGN VARIABLE BOUNDS, INITIAL VALUES AND SCALING FACTORS

FORMAT: 4F10

1	2	3	4	5	6	7	8
VLB	VUB	X	SCAL				

NOTE: READ ONE CARD FOR EACH OF THE NDV INDEPENDENT DESIGN VARIABLES.

FIELD	CONTENTS
1	VLB: Lower bound on the design variable. If VLB.LT.-1.0E+15, no lower bound.
2	VUB: Upper bound on the design variable. If VUB.GT.10.E+15, no upper bound.
3	X: Initial value of the design variable. If X is non-zero, this will supercede the value initialized by the user-supplied subroutine ANALIZ.
4	SCAL: Design variable scale factor. Not used if NSCAL.GE.0 in BLOCK C.

DATA BLOCK G   Omit if NDV=0 in block B

DESCRIPTION: DESIGN VARIABLE IDENTIFICATION

FORMAT: 2I10,F10

1	2	3	4	5	6	7	8
NDSGN	IDSGN	AMULT					

NOTE: READ ONE CARD FOR EACH OF THE NDVTOT DESIGN VARIABLES IN THE SAME ORDER AS IN BLOCK F.

FIELD	CONTENTS
1	NDSGN: Design variable number associated with this variable.
2	IDSGN: Global variable number associated with this variable.
3	AMULT: Constant multiplier on this variable. The value of the variable will be the value of the design variable, NDSGN, times AMULT. DEFAULT=1.0.

DATA BLOCK H   Omit IF=NDV 0 in block B

DESCRIPTION: NUMBER OF CONSTRAINED PARAMETERS

FORMAT: I10

1	2	3	4	5	6	7	8
NCONS							

FIELD	CONTENTS
1	NCONS: Number of constraint sets in the optimization problem.

REMARKS:

1) If two or more adjacent parameters in the global common block have the same limits imposed, these are part of the same constraint set.

DATA BLOCK: I   Omit if NDV=0 in block B, or NCONS=0 in block H

DESCRIPTION: CONSTRAINT IDENTIFICATION AND CONSTRAINT BOUNDS

FORMAT: 3I10

1	2	3	4	5	6	7	8
ICON	JCON	LCON					

NOTE: READ TWO CARDS FOR EACH OF THE NCONS CONSTRAINT SETS.

FIELD	CONTENTS
1	ICON: First global number corresponding to the constraint set.
2	JCON: Last global number corresponding to the constraint set. DEFAULT=ICON.
3	LCON: Linear constraint identifier for this constraint set. LCON=1 indicates linear constraints.

DATA BLOCK: I (CONTINUED)

FORMAT: 4F10

1	2	3	4	5	6	7	8
BL	SCAL1	BU	SCAL2				

FIELD	CONTENTS
1	BL: Lower bound on the constrained variables. If BL.LT.-1.0E+15, no lower bound.
2	SCAL1: Normalization factor on lower bound. DEFAULT=MAX of ABS(BL), 0.1.
3	BU: Upper bound on the constrained variables. If BU.GT.1.0E+15, no upper bound.
4	SCAL2: Normalization factor on upper bound. DEFAULT=MAX of ABS(BU), 0.1.

REMARKS:

1) The normalization factor should usually be defaulted.
2) The constraint functions sent to CONMIN are of the form: (BL-VALUE)/SCAL1 .LE. 0.0 and (VALUE - BU)/SCAL2 .LE. 0.0.
3) Each constrained parameter is converted to two con- constraints in CONMIN unless ABS(BL) or ABS(BU) exceeds 1.0E+15, in which case no constraint is created for that bound.

DATA BLOCK: P   Omit if NSV=0 in block B

DESCRIPTION: SENSITIVITY OBJECTIVES

FORMAT: 2I10

1	2	3	4	5	6	7	8
NSOBJ	IPSENS						

NOTE: TWO OR MORE CARDS ARE READ HERE.

FIELD	CONTENTS
1	NSOBJ: Number of separate objective functions to be calculated as function of the sensitivity variables.
2	IPSENS: Print control. If IPSENS.GT.0, detailed print will be called at each step in the sensitivity analysis. DEFAULT=No print.

DATA BLOCK: P (CONTINUED)

DESCRIPTION:

FORMAT: 8I10

1	2	3	4	5	6	7	8
NSN1	NSN2	NSN3	NSN4	....	....	....	
				.			

FIELD	CONTENTS
1-8	NSNI: Global variable number associated with the sensitivity objective functions.

REMARKS:

1) More than eight sensitivity objectives are allowed. Add data cards as required to contain data.

DATA BLOCK: Q Omit if NSV=0 in block B

DESCRIPTION: SENSITIVITY VARIABLES

FORMAT: 2I10

1	2	3	4	5	6	7	8
ISENS	NSENS						

NOTE: READ ONE SET OF DATA FOR EACH OF THE NSV SENSITIVITY VARIABLES. TWO OR MORE CARDS ARE READ FOR EACH SET OF DATA.

FIELD	CONTENTS
1	ISENS: Global variable number associated with the sensitivity variable.
2	NSENS: Number of values of this sensitivity variable to be read on the next card.

DATA BLOCK: Q (CONTINUED)

DESCRIPTION: 8F10

FORMAT:

1	2	3	4	5	6	7	8
SNS1	SNS2	SNS3	SNS4	...	...	...	...

FIELD	CONTENTS
1-8	SENSI: Values of the sensitivity variable. I=1,NSENS. I=1 corresponds to the nominal value.

REMARKS:

1) More than eight values of the sensitivity variable are allowed. Add data cards as required to contain the data.

DATA BLOCK: V

DESCRIPTION: COPES DATA 'END' CARD

FORMAT: 3A1

1	2	3	4	5	6	7	8
END							

FIELD	CONTENTS
1	The word 'END' in columns 1-3.

REMARKS

1) This card MUST appear at the end of the COPES data.
2) This ends the COPES input data.
3) Data for the user-supplied routine, ANALIZ, follows this.

HEAT PIPE ANALYSIS

Data for the heat pipe analysis follows the 'END' card in the COPES data deck. If the general design capability of COPES/CONMIN is not needed, the heat pipe analysis can be run by setting NCALC = 1 in field number 1 of data block B; or in a stand-alone mode by using the following main program.

```
C   MAIN PROGRAM FOR HEAT PIPE ANALYSIS
C   READ, EXECUTE, AND PRINT RESULTS
    DO 10 ICALC = 1,3
10  CALL ANALIZ (ICALC)
    STOP
    END
```

DATA BLOCK: AA

DESCRIPTION: ELEMENT CONNECTIVITY

FORMAT: 3I5

1	2	3	4	5	6	7	8
NEL	NSNP	NBAN					

FIELD	CONTENTS
1	NEL: Number of elements
2	NSNP: Number of system nodal points
3	NBAN: System band width

DATA BLOCK: BB

DESCRIPTION: ELEMENT CONNECTIVITY

FORMAT: 4I5 I = 1,3; IEL=1, NEL

1	2	3	4	5	6	7	8
IEL	ICOR(IEL,I)						

FIELD	CONTENTS
1	IEL: The element number
2	ICOR(IEL,I): System nodal point corresponing to nodal point I of element IEL

REMARKS:

1) Number all elements with convective boundaries first from top to bottom, then number the remaining elements.
2) Number the nodal points of each element moving in a counter-clockwise direction.
3) The elements with convective boundaries have nodal points 1 and 2 located on the convective boundary.

DATA BLOCK: CC

DESCRIPTION: CONDENSER GEOMETRY

FORMAT: 7G10.5

1	2	3	4	5	6	7	8
CLI	CANGL	RBASEI	R2I	THICKI	BFIN	TZ	

FIELD	CONTENTS
1	CLI: Condenser length (inches)
2	CANGL: Cone half angle (degrees)
3	RBASEI: Inside radius of condenser base (inches)
4	R2I: Intermediate radius (inches)
5	THICKI: Condenser wall thickness (inches)
6	BFIN: Height of fin (inches)
7	TZ: Nodal point temperature initial guess (degrees F)

REMARKS

1) Set TZ equal to TSS-Saturation temperature of the working fluid

DATA BLOCK: DD

DESCRIPTION: FINITE ELEMENT GEOMETRY

FORMAT: 5I5

1	2	3	4	5	6	7	8
NDIV	NEST	NEFB	NBOTI	NBOTF			

FIELD	CONTENTS
1	NDIV: Number of increments along the length of the condenser.
2	NEST: Number of the element on the right end of the trough.
3	NEFB: The element number with convective boundary located at the base of the fin.
4	NBOTI: The element number with convective boundary located at the right hand of the bottom side.
5	NBOTF: The element number with convective boundary located at the left hand of the bottom side.

DATA BLOCK: EE

DESCRIPTION: DATA FOR RUNNING

FORMAT: 4F10.2

1	2	3	4	5	6	7	8
RPM	TSS	TINF	HINF				

FIELD	CONTENTS
1	RPM: Rotation rate of heat pipe (RPM)
2	TSS: Saturation temperature of the working fluid (degrees F)
3	TINF: Outside temperature (degrees F)
4	HINF: Outside convective heat transfer coefficient (BTU/HR·FT^2°F)

DATA BLOCK: FF

DESCRIPTION: CONVERGENCE CRITERION

FORMAT: G10.9

1	2	3	4	5	6	7	8
CRIT							

FIELD	CONTENTS
1	CRIT: Convergence criterion on finite element solution for incremental heat transfer rate

DATA BLOCK: GG

DESCRIPTION: INTERNAL FIN GEOMETRY

FORMAT: 2G10.5

1	2	3	4	5	6	7	8
FANGL	ZOA						

FIELD	CONTENTS
1	FANGL: Fin half angle (degrees)
2	ZOA: Ratio of trough width to fin base width

DATA BLOCK: HH

DESCRIPTION: INTERNAL FIN GEOMETRY

FORMAT: I5

1	2	3	4	5	6	7	8
IFF							

FIELD	CONTENTS
1	IFF: (n-1), where n is the number of rows of the upper triangular fin section

NOTE: See Figure A-1.

DATA BLOCK: II

DESCRIPTION: INTERNAL FIN GEOMETRY

FORMAT: 16I5 I=1,IFF

1	2	3	4	5	6	7	8
KFIN(I)	KFF(I)						
			.				

FIELD	CONTENTS
1	KFIN: The number of system nodal points located on the symmetric boundary of triangular fin section, but does not include the system nodal points located at the base of the fin and the apex.
2	KFF: The number of system nodal points located along the fin convective boundary, but does not include the system nodal points located at the base of the fin and the apex.

NOTE: See Figure A-1.

DATA BLOCK: JJ

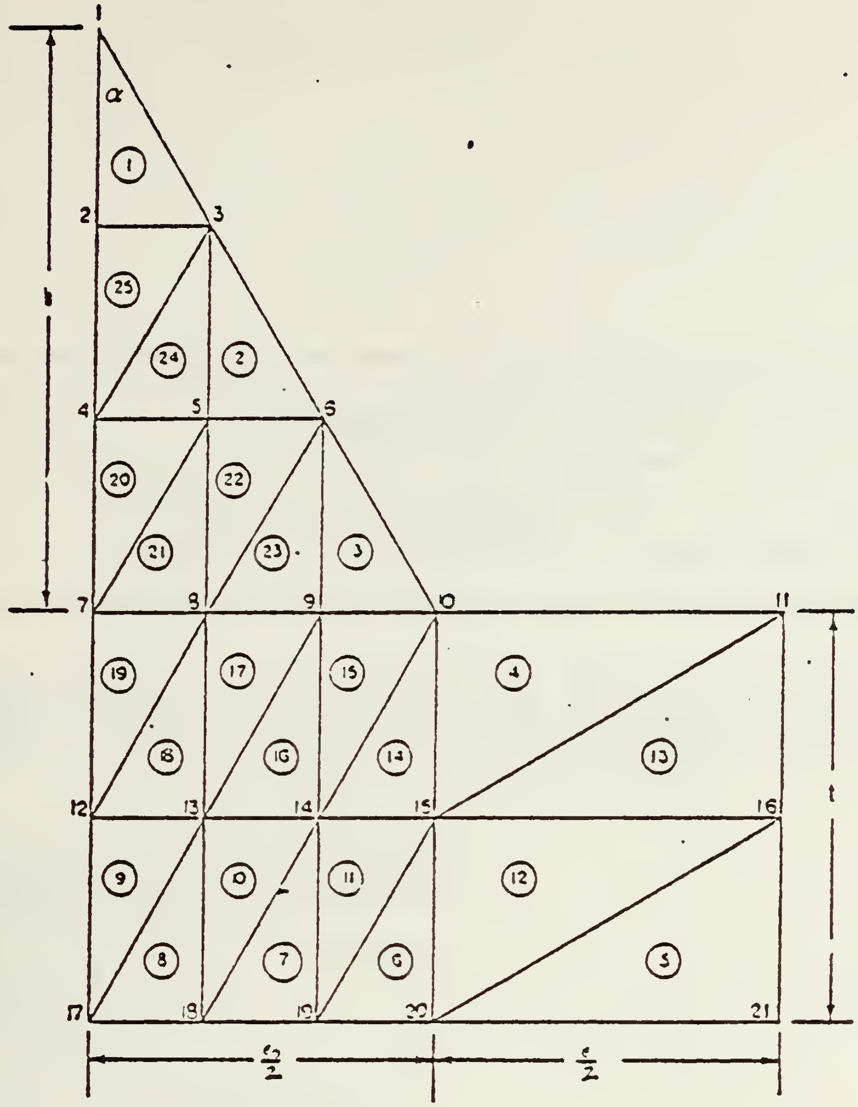
DESCRIPTION: INTERNAL FIN GEOMETRY

FORMAT: 2G10.5,4I5

1	2	3	4	5	6	7	8
DOBF	DOTH	JTC	JLC	JINT	KT		

FIELD	CONTENTS
1	DOBF: Number of column within fin.
2	DOTH: Number of column within though.
3	JTC: The number of the system nodal point located at the junction of the symmetry boundary and the line of intersection between the fin and the condenser wall.
4	JLC: The number of the system nodal point located at the center of system coordinates.
5	JINT: The numerical difference between the two adjacent system nodal points vertically at the condenser section.
6	KT: The number of rows within the wall section.

NOTE: See Figure A-1.



IFF = 2		DOBF = 3.0		JLC = 17			
KFIN = 2 ; 4		DOTH = 1.0		JINT = 5			
KFF = 3 ; 6		JTC = 7		KT = 2			

Figure A-1. Specification for Input Data to Determine the Coordinates of the System Nodal Points

APPENDIX B

COMPUTER PROGRAM LISTING

```
C     *****************************************************************
C     *                                                               *
C     *    ANALYSIS CF ROTATING HEAT PIPE , USING TRIANGULAR          *
C     *                   ELEMENT MODEL                               *
C     *    COMPILED BY MAJOR IGNATIUS.S.PURNOMO IN JUNE 1978          *
C     *                                                               *
C     *        MODIFIED TO PERMIT NUMERICAL OPTIMIZATION              *
C     *                 USING COPES/CONMIN                            *
C     *      BY LCDR WILLIAM A. DAVIS, JR. IN SEPTEMBER 1980          *
C     *                                                               *
C     *****************************************************************
C
C
C         SUBROUTINE ANALIZ CHANGES THE ARRAY OF DESIGN
C         VARIABLES FROM SINGLE TO DOUBLE PRECISION AND
C         BACK.  COPES/CONMIN USES SINGLE PRECISION ONLY;
C         DOUBLE PRECISION IS MAINTAINED IN SUBROUTINE FUN
C         TO ALLOW FOR POSSIBLE ILL CONDITIONING.
C
C
          SUBROUTINE ANALIZ (ICALC)
          COMMON /GLCBCM/ ARRAY(750)
          COMMON /GLCB1/ BARAY(50)
          REAL*8 BARAY
          IF (ICALC.GT.1) GO TO 20
          DO 10 I=1,50
          BARAY(I)=0.0D0
       10 CONTINUE
       20 CONTINUE
          DO 30 I=1,50
       30 BARAY(I)=DBLE(ARRAY(I))
          CALL FUN (ICALC)
          DO 40 I=1,50
          ARRAY(I)=SNGL(BARAY(I))
       40 CONTINUE
          RETURN
          END
C         GUIDE TO FORTRAN VARIABLE NAMES
C
C      ALFA     FIN HALF ANGLE (RADIANS)
C      BFIN     HEIGHT OF FIN (INCHES)
C      BVIN     HEIGHT OF FIN (FEET)
C      CALFA    COSINE OF ALFA
C      CANGL    CONE HALF ANGLE (DEGREES)
C      CBASE    INSIDE CIRCUMFERENCE OF CONDENSER (FEET)
C      CEXIT    INSIDE CIRCUMFERENCE AT CONDENSER EXIT (FEET)
C      CF       THERMAL CONDUCTIVITY OF CONDENSATE FILM (BTU/HR)
C      CL       CONDENSER LENGTH (FEET)
C      CLI      CONDENSER LENGTH (INCHES)
C      CPHI     COSINE OF PHI
C      CRIT     CONVERGENCE CRITERION
```

```
C     DIV       FLOATING POINT VALUE OF NDIV
C     DMTOT     CONDENSATE MASS FLOW RATE
C     DOBF      NUMBER OF COLUMN WIWTHIN THE FIN
C     DOTH      NUMBER OF COLUMN WITHIN THE TROUGH
C     FANGL     FIN HALF ANGLE (DEGREES)
C     H         CONVECTIVE HEAT TRANSFER COEFFICIENT (BTU/HR FT2)
C     HFG       LATENT HEAT OF VAPORIZATION (BTU/LBM)
C     IEL       THE ELEMENT NUMBER                        .
C     JLC       NUMBER OF SYSTEM NODAL POINT LOCATED AT
C               THE CENTER OF SYSTEM COORDINATE
C     JTC       NUMBER OF SYSTEM NODAL POINT LOCATED AT
C               THE JUNCTION OF THE SYNNETRY BOUNDARY AND
C               THE LINE OF INTERSECTION BETWEEN THE FIN
C               AND THE CONDENSER WALL
C     KFF       NUMBER OF SYSTEM NODAL POINTS LOCATED ALONG
C               THE FIN CONVECTIVE BOUNDARY
C     KFIN       NUMBER OF SYSTEM NODAL POINTS LOCATED ON THE
C               SYSMMETRIC BOUNDARY OF TRIANGULAR FIN SECTION
C               NOT COUNTING POINTS AT BASE AND APEX
C     KT        NUMBER OF ROWS WITHIN THE WALL SECTION
C     NBAN      SYSTEM BAND WIDTH
C     NBOTF     LAST ELEMEMT AT BOTTOM SIDE
C     NBOTI     FIRST ELEMENT AT BOTTOM SIDE
C     NDIV      NUMBER OF INCREMENT
C     NEFB      ELEMENT NUMBER AT BASE OF FIN
C     NEL       NUMBER OF ELEMENTS
C     NEST      ELEMENT NUMBER AT END OF TROUGH
C     NSNP      NUMBER OF SYSTEM NODAL POINTS
C     PHI       CONE HALF ANGLE (RADIANS)
C     PI        PI
C     RBASE     INSIDE RADIUS OF CONDENSER BASE (FEET)
C     RBASEI    INSIDE RADIUS OF CONDENSER BASE (INCHES)
C     REXIT     INSIDE RADIUS OF CONDENSER EXIT (FEET)
C     SALFA     SINE OF ALFA
C     SPHI      SINE OF PHI
C     THICK     CONDENSER WALL THICKNESS (FEET)
C     THICKI    CONDENSER WALL THICKNESS (INCHES)
C     TPHI      TANGENT OF PHI
C     ZFIN      NUMBER OF FINS
C     ZOA       RATIO OF TROUGH WIDTH TO FIN BASE WIDTH
C
C
C        SUBROUTINE FUN READS INPUT DATA, PERFORMS HEAT
C        TRANSFER ANALYSIS, AND PRINTS RESULTS.
C
C
```

```
      SUBROUTINE FUN (ICALC)
      IMPLICIT REAL*8(A-H,O-Z)
      COMMON /GLOB1/ CLI,CANGL,RBASEI,R2I,THICKI,BFIN,TZ,TSS
     1,TINF,HINF,FANGL,ZOA,ZFIN,BOA,QTOT,RPM
      DIMENSION Z(200),EPS(200),HZ(200),XCOF(5),COF(5),T(200
     1),QB(200),DMDOT(200),UF(200),CF(200),CW(200),AMTOT(200
     2),R(200),CINC(200),TB(200),TT(200),TIB(200),QTINC(200)
     3,NC(200),QTOTAL(100),TE(200),ROOTR(4),ROOTI(4),DEL(200
     4),TBM(200),RHOF(200)
      COMMON /APOL/ DOBF,DOTH,KFIN(50),KFF(50),IFF,JTC,JLC,J
     1INT,KT
      COMMON /MAFO/ A(200,50),F(200,1),H(200),TS(200),TSAT,C
     1K,NEL,NSNP,NBAN,ICOR(200,3)
      COMMON /PCRD/ X(200),Y(200),EZERO,BVIN,THICK,TALFA,APS
      HDEN(A1,B1,ZZ)=(-1.0D0*(A1*ZZ**3/3.0D0+B1*ZZ**2/2.0D0)
     1)
C     IF ICALC=1 READ INPUT DATA
      IF (ICALC.GT.1) GO TO 10
C
C
C     *****    INPUT MODE    *****
C
C
C           ELEMENT CONNECTIVITIES
C
      READ (5,420) NEL,NSNP,NBAN
      WRITE (6,430) NEL,NSNP,NBAN
      READ (5,440) (IEL,(ICOR(IEL,I),I=1,3),IEL=1,NEL)
      WRITE (6,450)
      WRITE(6,251)(IEL,(ICOR(IEL,I),I=1,3),IEL=1,NEL)
C
C
C           THE CONDENSER GEOMETRY
C
      READ (5,460) CLI,CANGL,RBASEI,R2I,THICKI,BFIN,TZ
      WRITE (6,470) CLI,CANGL,RBASEI,R2I,THICKI,BFIN,TZ
      READ (5,480) NDIV,NEST,NEFB,NBOTI,NBOTF
      WRITE (6,490) NDIV,NEST,NEFB,NBOTI,NBOTF
C
C           DATA FOR RUNNING
C
      READ (5,500) RPM,TSS,TINF,HINF
      WRITE (6,510) RPM,TSS,TINF,HINF
C
C           THE CONVERGENCE CRITERIAN
C
      READ (5,520) CRIT
      WRITE (6,530) CRIT
C
C           INTERNAL FIN GEOMETRY
C
      READ (5,540) FANGL,ZOA
      WRITE (6,550) FANGL,ZOA
      READ (5,560) IFF
      WRITE (6,570) IFF
      READ (5,580) (KFIN(I),KFF(I),I=1,IFF)
      READ (5,590) DOBF,DOTH,JTC,JLC,JINT,KT
      NHB=NEFB/2
      NTM=NBOTI+(NBOTF-NBOTI)/2
      NBF=NBOTF+1
      WRITE (6,600) ICOR(NBOTI,2),ICOR(NEFB,1),ICOR(NTM,2),I
     1COR(NEST,1),ICOR(NBOTF,1)
      RETURN
C     IF ICALC=2 PERFORM THE HEAT TRANSFER ANALYSIS
   10 IF (ICALC.GT.2) GO TO 360
C
C
C     *****    EXECUTION MODE    *****
```

```
C
C
C     CONVERT UNITS OF ALL DIMENSIONAL PARAMETERS
C     TO FEET.  CONVERT UNITS OF ANGLES TO RADIANS.
      CL=CLI/12.0D0
      R2=R2I/12.0D0
      RBASE=RBASEI/12.0D0
      BVIN=BFIN/12.0D0
      DIV=DFLOAT(NDIV)
      PI=3.14159265358979D0
      PHI=2.0D0*CANGL*PI/360.0D0
      SPHI=DSIN(PHI)
      CPHI=DCOS(PHI)
      TPHI=DTAN(PHI)
      DELX=CL/DIV
      CBASE=2.0D0*PI*RBASE
      REXIT=RBASE+CL*TPHI
      CEXIT=2.0D0*PI*REXIT
      THICK=THICKI/12.0D0
      ALFA=FANGL*2.0D0*PI/360.0D0
      SALFA=DSIN(ALFA)
      CALFA=DCOS(ALFA)
      TALFA=DTAN(ALFA)
      EZERO=2.0D0*BVIN*TALFA
C
C             BOUNDARY CONDITIONS AND TEMPERATURE ESTIMATES
C             ALONG THE FIN BOUNDARY
C
      DO 20 NTINF=NBOTI,NBOTF
   20 TS(NTINF)=TINF
      DO 30 NNT=NBF,NEL
      TS(NNT)=0.0D0
   30 H(NNT)=0.0D0
      DO 40 IGT=1,NEST
      IE=ICOR(IGT,2)
   40 T(IE)=TZ
      IG=ICOR(NEST,1)
      T(IG)=TZ
      OMEGA=RPM*2.0D0*PI*60.0D0
      DO 50 KL=NBOTI,NBOTF
   50 H(KL)=HINF
      HIFN=HINF
      TSAT=TSS
      EPSO=ZOA*EZERO
      EOA=BVIN/(EZERO/2.0D0)
      ZFIN=CBASE/(EZERO+EPSO)
      SURFAR=ZFIN*(2.0D0*(BVIN/CALFA)+EPSO)
      EPSEX=(CEXIT-(ZFIN*EZERO))/ZFIN
      BETA=(EPSEX-EPSO)/DIV
      ZZERO=BVIN/CALFA
      ZA=0.0D0
      DO 60 NSAT=1,NEST
   60 TS(NSAT)=TSAT
      TSOLID=(TSAT+TINF)/2.0D0
C       TEMPORARY CHANGE - TFILM
      QT=0.0D0
      QBTOT=0.0D0
      QT1=0.0D0
      QTF=0.0D0
      QTOT=0.0D0
      DMTOT=0.0D0
      NK=NDIV+1
      DO 350 NI=1,NK
      R(NI)=R2+NI*DELX*SPHI
      EPS(NI)=EPSO+NI*BETA
      APS=EPS(NI)
C
```

```
C           NCCAL POINT COORDINATES
C
      CALL COORD
      Z(1)=ZA
      DO 70 IZEL=1,NEFB
      NA=ICOR(IZEL,1)
      NB=ICOR(IZEL,2)
      XE=X(NA)-X(NB)
      YE=Y(NA)-Y(NB)
      ELZ=DSQRT(XE**2+YE**2)
   70 Z(IZEL+1)=Z(IZEL)+ELZ
      XZB=X(ICCR(NHB,1))-X(ICOR(1,2))
      YZB=Y(ICCR(NHB,1))-Y(ICOR(1,2))
      ZB=DSQRT(XZB**2+YZB**2)
      ZC=ZZERO
      IM=1
C
C        PARABOLIC TEMPERATURE DISTRIBUTION ALONG THE FIN
C        BOUNDARY,USING LAGRANGE INTERPOLATION
C
   80 TP1=T(ICCR(1,2))
      TP2=T(ICOR(NHB,1))
      TP3=T(ICCR(NEFB,1))
      AP1=TP1/(ZB*ZC)
      AP2=TP2/(ZB*(ZB-ZC))
      AP3=TP3/(ZC*(ZC-ZB))
      BP1=-(ZB+ZC)*AP1
      BP2=-ZC*AP2
      BP3=-ZB*AP3
      A1=AP1+AP2+AP3
      B1=BP1+BP2+BP3
      TC=0.0D0
      DO 90 NY=1,NEST
   90 TC=TC+T(ICCR(NY,2))
      AY=DFLOAT(NY+1)
      TF=(TC+T(ICCR(NY,1))+AY*TS(NY))/(2.0D0*AY)
C
C           SOLID-FLUID PROPERTIES
C
      HFG=1097.2D0-0.601875D0*TS(1)
      RHOF(NI)=62.774D0-0.00255698D0*TF-0.00005357200*TF**2
      CF(NI)=0.3034D0+0.000738927D0*TF-0.0000014732100*TF**2
      UF(NI)=0.001397D0-0.000014669D0*TF+0.0000000631253D0*T
     1F**2-0.000000000097656 9D0*TF**3
      LF(NI)=36C0*UF(NI)
C     CW(NI)=231.7772D0-0.02222D0*TSOLID
      CW(NI)=8.776+0.00265D0*TSOLID
C     CW(NI)=1.0
C     CW(NI)=20000.0
      CK=CW(NI)
      CONST=RHOF(NI)**2*OMEGA**2*HFG*CPHI*CALFA*R(NI)
C
C     INITIAL FILM THICKNESS
C     DEL(1)=0.0000675200
      IF (NI.GT.1) GO TO 100
      DEL(1)=1.1C7*(((TSAT-TINF)*CF(NI)/(UF(NI)*HFG*3600.0D0
     1))**0.25)*((UF(NI)/(RHOF(NI)*OMEGA))**0.5)
  100 CONTINUE
C
C           AVERAGE ELEMENT CONVECTIVE COEFFICIENT ALONG
C           THE FIN BOUNDARY
C
      ZSTAR=ZZERC-DEL(NI)/CALFA
      AZZ=DEL(NI)/SALFA
      ZZ=ZSTAR
      AZS=DABS(4*CF(NI)*UF(NI)*HDEN(A1,B1,ZZ)/CONST)**0.25D0
      HAC=0.0D0
```

```
      DO 190 IEL=1,NEFB
      AZ=Z(IEL)
      BZ=Z(IEL+1)
      IF (ZSTAR.LE.BZ) GO TO 110
      GO TO 120
  110 IF (HAC.NE.0.0D0) GO TO 180
      BZ=ZSTAR
  120 IF (IEL.NE.1) GO TO 130
      AK=(BZ-AZ)/5.0D0
      ZZ=AK
      GO TO 140
  130 AK=(BZ-AZ)/4.0D0
      ZZ=AZ
  140 ZEL=4*AK
      DO 150 NH=1,5
      HZ(NH)=DABS(CF(NI)**3*CONST/(4*UF(NI)*HDEN(A1,B1,ZZ)))
     1**0.25D0
  150 ZZ=ZZ+AK
      CONH=AK*(HZ(1)+4*HZ(2)+2*HZ(3)+4*HZ(4)+HZ(5))/(3*ZEL)
      IF (ZSTAR.EQ.BZ) GO TO 160
      H(IEL)=CONH
      GO TO 190
  160 AZ=ZSTAR
      HAZ=CONH*(AZ-Z(IEL))
      DELA=AZS
  170 BZ=Z(IEL+1)
      DELB=(BZ-ZSTAR)*AZZ/(ZZERO-ZSTAR)
      DELZ=(DELA+DELB)/2.0D0
      HAC=(BZ-AZ)*CF(NI)/DELZ
      H(IEL)=(HAZ+HAC)/(BZ-Z(IEL))
      GO TO 190
  180 AZ=Z(IEL)
      DELA=DELB
      HAZ=0.0D0
      GO TO 170
  190 CONTINUE
      NETI=NEFB+1
      DO 200 IEL=NETI,NEST
  200 H(IEL)=CF(NI)/DEL(NI)
C
C             ENTRY INTO THE FINITE ELEMENT SOLUTION
C
      CALL FORMAF
      CALL BANDEC (NSNP,NBAN,1)
C
C             THE TEMPERATURE DISTRIBUTION
C
      DO 210 NT=1,NSNP
  210 T(NT)=F(NT,1)
      TIB(NI)=T(ICOR(NBOTI,2))
      TT(NI)=T(ICOR(NEFB,1))
      TBM(NI)=T(ICOR(NTM,2))
      TE(NI)=T(ICOR(NEST,1))
      TB(NI)=T(ICOR(NBOTF,1))
      TTS=0.0D0
      DO 220 NS=1,NSNP
  220 TTS=TTS+T(NS)
      PN=DFLOAT(NS)
      TSOLID=TTS/PN
C
C             Q AT THE BOTTOM SIDE
C
      QBI=0.0DC
      DO 230 IBEL=NBOTI,NBOTF
      NKA=ICOR(IBEL,1)
      NKB=ICOR(IBEL,2)
      XB=X(NKA)-X(NKB)
```

```
      YB=Y(NKA)-Y(NKB)
      ELB=DSQRT(XB**2+YB**2)
  230 QBI=QBI+(T(NKA)+T(NKB)-2*TS(IBEL))*ELB*H(IBEL)/2.0D0
      QB(NI)=QBI*DELX
C
C             ITERATION UNTIL CONVERGENCE CRITERIA IS MET
C
      IF (IM.EQ.1) GO TO 240
      QJ=QBI
      GO TO 250
  240 QI=QBI
      IM=2
      GO TO 80
  250 AQ=DABS(QJ-QI)/QJ
      IF (AQ.LE.CRIT) GO TO 260
      QI=QJ
      GO TO 80
  260 DMDOT(NI)=2.*QBI*DELX/HFG
      DMTOT=DMTOT+DMDOT(NI)
      C1=RHOF(NI)**2*OMEGA**2*R(NI)*SPHI/(3*UF(NI))
      XCOF(1)=-DMTOT
      XCOF(2)=0.0D0
      XCOF(3)=0.0D0
      XCOF(4)=C1*EPS(NI)
      XCOF(5)=C1*TALFA
      M=4
      CALL DPOLRT (XCOF,COF,M,ROOTR,ROOTI,IER)
      IF (ROOTR(1).GT.0.0D0) GO TO 270
      IF (ROOTR(2).GT.0.0D0) GO TO 280
      IF (ROOTR(3).GT.0.0D0) GO TO 290
      IF (ROOTR(4).GT.0.0D0) GO TO 300
      WRITE (6,610)
      WRITE (6,620) (ROOTR(I),I=1,4)
C     GO TO 6100
C
C             THE CONDENSATE THICKNESS
C
  270 DEL(NI+1)=ROOTR(1)
      GO TO 310
  280 DEL(NI+1)=ROOTR(2)
      GO TO 310
  290 DEL(NI+1)=ROOTR(3)
      GO TO 310
  300 DEL(NI+1)=ROOTR(4)
  310 QEL=0.0D0
      IF (NI.NE.1) GO TO 320
C
C             Q FROM THE TOP SIDE
C
C     Q THROUGH FIN
C
  320 DO 330 IQEL=1,3
      KA=ICOR(IQEL,1)
      KB=ICOR(IQEL,2)
      XQEL=X(KA)-X(KB)
      YQEL=Y(KB)-Y(KA)
      ELM=DSQRT(XQEL**2+YQEL**2)
      QEL=QEL+(2*TS(IQEL)-T(KA)-T(KB))*ELM*H(IQEL)/2.0D0
  330 CONTINUE
      QINC(NI)=QEL*DELX
      AMTOT(NI)=DMTOT
      QET=QEL*DELX*ZFIN*2
      QT=QT+QET
      QA=QBI*DELX*ZFIN*2
      QTOT=QTOT+QA
C
C     Q THROUGH TROUGH
```

```
C
      DO 340 IQEL=4,4
      KA=ICOR(ICEL,1)
      KB=ICOR(IQEL,2)
      XQEL=X(KA)-X(KB)
      YQEL=Y(KB)-Y(KA)
      ELM=DSQRT(XQEL**2+YQEL**2)
      CTRF=(2*TS(IQEL)-T(KA)-T(KB))*ELM*H(IQEL)/2.0D0
  340 CONTINUE
      QTINC(NI)=QTRF*DELX
      CTOTAL(NI)=CINC(NI)+QTINC(NI)
      QTRFT=QTRF*DELX*ZFIN*2.
      CTF=QTF+QTRFT
  350 CONTINUE
      RETURN
  360 CONTINUE
C
C
C     *****   CUTPUT MODE   *****
      WRITE (6,630)
      CO 370 NR=1,NDIV
  370 WRITE (6,640) NR,QINC(NR),CTINC(NR),QTOTAL(NR)
      WRITE (6,650) QT,QTF
      WRITE (6,660) CLI,CANGL,RBASEI,R2I,THICKI,BFIN,RPM,TSS
     1,TINF,HINF,CRIT,FANGL,ZOA,IFF
      WRITE (6,670) BOA,ZOA,ZFIN,BVIN,SURFAR
      WRITE (6,680)
      DO 380 NP=1,NSNP
  380 WRITE (6,690) NP,X(NP),Y(NP),T(NP)
      WRITE (6,700)
      CO 390 KKL=1,NBOTF
      NKX=ICOR(KKL,1)
      NKY=ICOR(KKL,2)
      XP=X(NKX)-X(NKY)
      YP=Y(NKX)-Y(NKY)
      EXY=DSQRT(XP**2+YP**2)
      CEP=DABS((T(NKX)+T(NKY)-2*TS(KKL))*EXY*H(KKL)/2.0D0)
      CEP=QEP*CELX
  390 WRITE (6,710) KKL,H(KKL),EXY,QEP
      WRITE (6,720) CRIT
      WRITE (6,730) HFG,ZFIN,H(NBOTF),TSAT,RPM,QTOT,QT,FANGL
      WRITE (6,740)
      DO 400 NR=1,NDIV
  400 WRITE (6,75C) NR,DEL(NR),QB(NR),AMTOT(NR),TIB(NR),TT(N
     1R),TE(NR),TB(NR)
      WRITE (6,760)
      DO 410 NG=1,NDIV,2
  410 WRITE (6,770) NG,CW(NG),CF(NG),RHOF(NG),UF(NG),EPS(NG)
     1,R(NG),TBM(NG),QINC(NG)
      RETURN
C
  420 FORMAT (3I5)
  430 FORMAT (/2X,15HNO.OF.ELEMENTS=,I5,10X,34HNO.OF.SYSTEM
     1                 N.P.=,I5,10X,13HNO.OF BANDED=,I5)
  440 FORMAT (4I5)
  450 FORMAT (/2X,7HELEMENT,10X,3HNP1,14X,3HNP2,15X,3HNP3)
  460 FORMAT (7G10.5)
  470 FORMAT (4X,5HCLI= ,E12.5,/,4X,7HCANGL= ,E12.5,/,4X,8HR
     1BASEI.=,E12.5,/,4X,5HR2I= ,E12.5,/,4X,8HTHICKI= ,E12.5
     2,/,4X,6HBFIN= ,E12.5,/,4X,4HTZ= ,E12.5)
  480 FORMAT (5I5)
  490 FORMAT (4X,6HNDIV= ,I10,/,4X,6HNEST= ,I10,/,4X,6HNEFB=
     1 ,I10,/,4X,7HNBOTI= ,I10,/,4X,7HNBOTF= ,I10)
  500 FORMAT (4F10.2)
  510 FORMAT (4X,5HRPM= ,E12.5,/,4X,5HTSS= ,E12.5,/,4X,6HTIN
     1F= ,E12.5,/,4X,6HHINF= ,E12.5)
  520 FORMAT (G10.9)
```

```
530  FORMAT (4X,6HCRIT= ,E12.5)
540  FORMAT (2G10.5)
550  FORMAT (4X,7HFANGL= ,E12.5,/,4X,5HZOA= ,E12.5)
560  FORMAT (I5)
570  FORMAT (4X,5HIFF= ,I10)
580  FORMAT (16I5)
590  FORMAT (2G10.5,4I5)
600  FORMAT (///5X,4HTIB=,I5,10X,3HTT=,I5,/,5X,4HTBM=,I5,10
    1X,3HTE=,I5,/,6X,3HTB=,I5)
610  FORMAT (//10X,17HCRASH,CRASH,CRASH)
620  FORMAT (//5X,4(E12.7,3X))
630  FORMAT (2X,7HELEMENT,2X,4HQFIN,17X,7HQTROUGH,15X,6HQTO
    1TAL)
640  FORMAT (4X,I5,E12.5,10X,E12.5,10X,E12.5)
650  FORMAT (//,4X,11HQFIN TOTAL=,E12.5,10X,15HQTROUGH TOTA
    1L= ,E12.5)
660  FORMAT (/////,4X,5HCLI= ,E12.5,5X,7HCANGL= ,E12.5,/,4X
    1,8HRBASEI= ,E12.5,2X,5HR2I= ,E12.5,/,4X,8HTHICKI= ,E12
    2.5,2X,6HBFIN= ,E12.5,/,4X,5HRPM= ,E12.5,5X,5HTSS= ,E12
    3.5,/,4X,6HTINF= ,E12.5,4X,6HHINF= ,E12.5,/,4X,6HCRIT=
    4,E12.5,4X,7HFANGL= ,E12.5,/,4X,5HZOA= ,E12.5,5X,5HIFF=
    5 ,I10)
670  FORMAT (1H1,//2X,4HBOA=,G12.5,5X,4HZOA=,G12.5,5X,5HZFI
    1N=,G12.5,5X,5HBFIN=,G12.5,5X,13HSURFACE AREA=,G12.5)
680  FORMAT (//5X,2HNP,6X,1HX,12X,1HY,12X,1HT)
690  FORMAT (/2X,I3,3X,3(F10.6,3X))
700  FORMAT (//2X,2HEL,8X,1HH,11X,9HEL-LENGTH,15X,4HQ-EL)
710  FORMAT (/2X,I2,3X,E12.5,3X,E12.5,10X,E12.5)
720  FORMAT (/2X,22HCONVERGENCE CRITERIAN=,E15.8)
730  FORMAT (1H ,//,5X,4HHFG=,E12.5,/,5X,11HNO.OF FINS=,E12
    1.5,/,5X,6HH-OUT=,E12.5,/,5X,5HTSAT=,E12.5,/,5X,4HRPM=,
    2E12.5,/,5X,6HQ-BOT=,E12.5,/,5X,6HQFIN =,E12.5,/,5X,11H
    3HALF-ANGLE=,F8.3)
740  FORMAT (1H0,6X,1HJ,4X,14HFILM THICKNESS,6X,8HQ-INCREM,
    16X,8HMASS-TOT,7X,3HTIB,8X,2HTT,10X,2HTE,8X,2HTB)
750  FORMAT (1H ,4X,I4,4X,F12.10,4X,F10.4,6X,F9.5,6X,F5.1,6
    1X,F5.1,6X,F5.1,6X,F5.1)
760  FORMAT (1H0,6X,1HJ,6X,6HK-WALL,4X,6HK-FILM,3X,7HDENSIT
    1Y,4X,9HVISC-FILM,6X,7HEPSILON,5X,6HRADIUS,5X,3HTBM,5X,
    25HQ-BOT)
770  FORMAT (1H ,4X,I4,4X,F7.3,4X,F6.4,4X,F6.3,4X,F9.7,4X,F
    19.7,4X,F7.5,5X,F5.1,1X,1PD12.3)
     END
     SUBROUTINE COORD
     IMPLICIT REAL*8(A-H,O-Z)
     COMMON /GLOB1/ CLI,CANGL,RBASEI,R2I,THICKI,BFIN,TZ,TSS
    1,TINF,HINF,FANGL,ZOA,ZFIN,BOA,QTOT,RPM
     COMMON /PCRD/ X(200),Y(200),EZERO,BVIN,THICK,TALFA,APS
     COMMON /APOL/ DOBF,DOTH,KFIN(50),KFF(50),IFF,JTC,JLC,J
    1INT,KT
     DELH=BVIN/DOBF
     X(1)=0.0D0
     Y(1)=THICK+BVIN
     N=1
     DO 20 I=1,IFF
     ICA=KFIN(I)
     ICB=KFF(I)
     CBA=DFLOAT(ICB-ICA)
     AN=0.0D0
     DO 10 II=ICA,ICB
     X(II)=X(1)+N*AN*DELH*TALFA/CBA
     Y(II)=Y(1)-N*DELH
 10  AN=AN+1.0D0
 20  N=N+1
     AN=0.0D0
     ICD=ICB-ICA+1
     DO 50 J=JTC,JLC,JINT
```

```
      X(J)=X(1)
      Y(J)=(1.0D0-AN/DOTH)*THICK
      DO 30 JJ=1,ICD
      X(J+JJ)=X(J)+JJ*EZERO/(2*(CBA+1.0D0))
   30 Y(J+JJ)=Y(J)
      DO 40 K=1,KT
      X(J+JJ+K)=X(J+JJ)+K*APS/(2.0D0*KT)
   40 Y(J+JJ+K)=Y(J)
   50 AN=AN+1.0D0
      RETURN
      END
      SUBROUTINE FORMAF
      IMPLICIT REAL*8(A-H,O-Z)
      DIMENSION B(3),C(3),EA(3,3)
      COMMON /GLOB1/ CLI,CANGL,RBASEI,R2I,THICKI,BFIN,TZ,TSS
     1,TINF,HINF,FANGL,ZOA,ZFIN,BOA,QTOT,RPM
      COMMON /PCRD/ X(200),Y(200),EZERO,BVIN,THICK,TALFA,APS
      COMMON /MAFO/ A(200,50),F(200,1),H(200),TS(200),TSAT,C
     1K,NEL,NSNP,NBAN,ICOR(200,3)
      DO 20 N=1,NSNP
      F(N,1)=0.0D0
      DO 10 MA=1,NBAN
   10 A(N,MA)=0.0D0
   20 CONTINUE
      DO 70 IEL=1,NEL
      IA=ICOR(IEL,1)
      IB=ICOR(IEL,2)
      IC=ICOR(IEL,3)
      B(1)=Y(IB)-Y(IC)
      B(2)=Y(IC)-Y(IA)
      B(3)=Y(IA)-Y(IB)
      C(1)=X(IC)-X(IB)
      C(2)=X(IA)-X(IC)
      C(3)=X(IB)-X(IA)
      EL=DSQRT(C(3)**2+B(3)**2)
      AS=DABS((B(1)*C(2)-B(2)*C(1))/2.0D0)
      HC=H(IEL)/CK
      DO 60 J=1,3
      JJ=ICOR(IEL,J)
      DO 50 K=1,3
      KK=ICOR(IEL,K)
      EA(J,K)=(B(J)*B(K)+C(J)*C(K))/(4*AS)
      IF (HC.EQ.0.0D0) GO TO 40
      HEL=HC*EL/6.0D0
      IF (J.EQ.3) GO TO 40
      IF (K.EQ.3) GO TO 40
      IF (J.EQ.K) GO TO 30
      EA(J,K)=EA(J,K)+HEL
      GO TO 40
   30 EA(J,K)=EA(J,K)+2*HEL
   40 IF (KK.LT.JJ) GO TO 50
      NW=KK-JJ+1
      A(JJ,NW)=A(JJ,NW)+EA(J,K)
   50 CONTINUE
   60 CONTINUE
      FE=HC*TS(IEL)*EL/2.0D0
      F(IA,1)=F(IA,1)+FE
      F(IB,1)=F(IB,1)+FE
   70 CONTINUE
      RETURN
      END
      SUBROUTINE BANDEC (NEQ,MAXB,NVEC)
      IMPLICIT REAL*8(A-H,O-Z)
      COMMON /GLOB1/ CLI,CANGL,RBASEI,R2I,THICKI,BFIN,TZ,TSS
     1,TINF,HINF,FANGL,ZOA,ZFIN,BOA,QTOT,RPM
      COMMON /PCRD/ X(200),Y(200),EZERO,BVIN,THICK,TALFA,APS
      COMMON /MAFO/ A(200,50),F(200,1),H(200),TS(200),TSAT,C
```

```
     1K,NEL,NSNP,NBAN,ICOR(200,3)
      LOOP=NEQ-1
      DO 20 I=1,LCOP
      MB=I+1
      NB=MINO(I+MAXB-1,NEQ)
      DO 20 J=MB,NB
      L=J+2-MB
      D=A(I,L)/A(I,1)
      DO 10 MM=1,NVEC
   10 F(J,MM)=F(J,MM)-D*F(I,MM)
      MM=MINO(MAXB-L+1,NEQ-J+1)
      DO 20 K=1,MM
      NN=L+K-1
   20 A(J,K)=A(J,K)-D*A(I,NN)
      DO 30 I=1,NVEC
   30 F(NEQ,I)=F(NEQ,I)/A(NEQ,1)
      DO 50 I=2,NEQ
      J=NEQ-I+1
      K=MINO(NEQ-J+1,MAXB)
      DO 50 MM=1,NVEC
      DO 40 L=2,K
      MB=J+L-1
   40 F(J,MM)=F(J,MM)-A(J,L)*F(MB,MM)
   50 F(J,MM)=F(J,MM)/A(J,1)
      RETURN
      END
C     SUBROUTINE DPOLRT COMPUTES THE ROOTS OF A REAL
C     POLYNOMIAL USING A NEWTON-RAPHSON ITERATIVE
C     TECHNIQUE.
C
C
      SUBROUTINE DPOLRT (XCOF,COF,M,ROOTR,ROOTI,IER)
      IMPLICIT REAL*8 (A-H), REAL*8 (O-Z)
      DIMENSION XCOF(1),COF(1),ROOTR(1),ROOTI(1)
C
C
      IFIT=0
      N=M
      IER=0
      IF (XCOF(N+1)) 10,40,10
   10 IF (N) 20,20,60
C
C        SET ERRCR CODE TO 1
C
   20 IER=1
   30 RETURN
C
C        SET ERRCR CODE TO 4
C
   40 IER=4
      GO TO 30
C
C        SET ERRCR CODE TO 2
C
   50 IER=2
      GO TO 30
   60 IF (N-36) 70,70,50
   70 NX=N
      NXX=N+1
      N2=1
      KJ1=N+1
      DO 80 L=1,KJ1
      MT=KJ1-L+1
   80 COF(MT)=XCCF(L)
C
C        SET INITIAL VALUES
C
```

```
   90 XO=.00500101
      YO=0.01000101
C
C        ZERO INITIAL VALUE COUNTER
C
      IN=0
  100 X=XO
C
C        INCREMENT INITIAL VALUES AND COUNTER
C
      XO=-10.0*YO
      YO=-10.0*X
C
C        SET X AND Y TO CURRENT VALUE
C
      X=XO
      Y=YO
      IN=IN+1
      GO TO 120
  110 IFIT=1
      XPR=X
      YPR=Y
C
C        EVALUATE POLYNOMIAL AND DERIVATIVES
C
  120 ICT=0
  130 UX=0.0
      UY=0.0
      V=0.0
      YT=0.0
      XT=1.0
      U=COF(N+1)
      IF (U) 140,270,140
  140 DO 150 I=1,N
      L=N-I+1
      XT2=X*XT-Y*YT
      YT2=X*YT+Y*XT
      U=U+COF(L)*XT2
      V=V+COF(L)*YT2
      FI=I
      UX=UX+FI*XT*COF(L)
      UY=UY-FI*YT*COF(L)
      XT=XT2
  150 YT=YT2
      SUMSQ=UX*UX+UY*UY
      IF (SUMSQ) 160,230,160
  160 DX=(V*UY-U*UX)/SUMSQ
      X=X+DX
      DY=-(U*UY+V*UX)/SUMSQ
      Y=Y+DY
      IF (DABS(DY)+DABS(DX)-1.0E-05) 210,170,170
C
C        STEP ITERATION COUNTER
C
  170 ICT=ICT+1
      IF (ICT-500) 130,180,180
  180 IF (IFIT) 210,190,210
  190 IF (IN-5) 100,200,200
C
C        SET ERROR CODE TO 3
C
  200 IER=3
      GO TO 30
  210 DO 220 L=1,NXX
      MT=KJ1-L+1
      TEMP=XCOF(MT)
      XCOF(MT)=COF(L)
```

```
220 COF(L)=TEMP
    ITEMP=N
    N=NX
    NX=ITEMP
    IF (IFIT) 250,110,250
230 IF (IFIT) 240,100,240
240 X=XPR
    Y=YPR
250 IFIT=0
    IF (DABS(Y/X)-1.0E-04) 280,260,260
260 ALPHA=X+X
    SUMSQ=X*X+Y*Y
    N=N-2
    GO TO 290
270 X=0.0
    NX=NX-1
    NXX=NXX-1
280 Y=0.0
    SUMSQ=0.0
    ALPHA=X
    N=N-1
290 COF(2)=COF(2)+ALPHA*COF(1)
    DO 300 L=2,N
300 COF(L+1)=COF(L+1)+ALPHA*COF(L)-SUMSQ*COF(L-1)
310 ROOTI(N2)=Y
    ROOTR(N2)=X
    N2=N2+1
    IF (SUMSQ) 320,330,320
320 Y=-Y
    SUMSQ=0.0
    GO TO 310
330 IF (N) 30,30,90
    END
```

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c.1 Optimization of an
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